IN TWO SECTIONS—SECTION TWO

TRANSACTIONS

of The American Society of Mechanical Engineers

SOCIETY RECORDS—Part II

(Part I of Society Records for the year 1937, the Membership List, was issued as Section Two of the Transactions for February, 1937)

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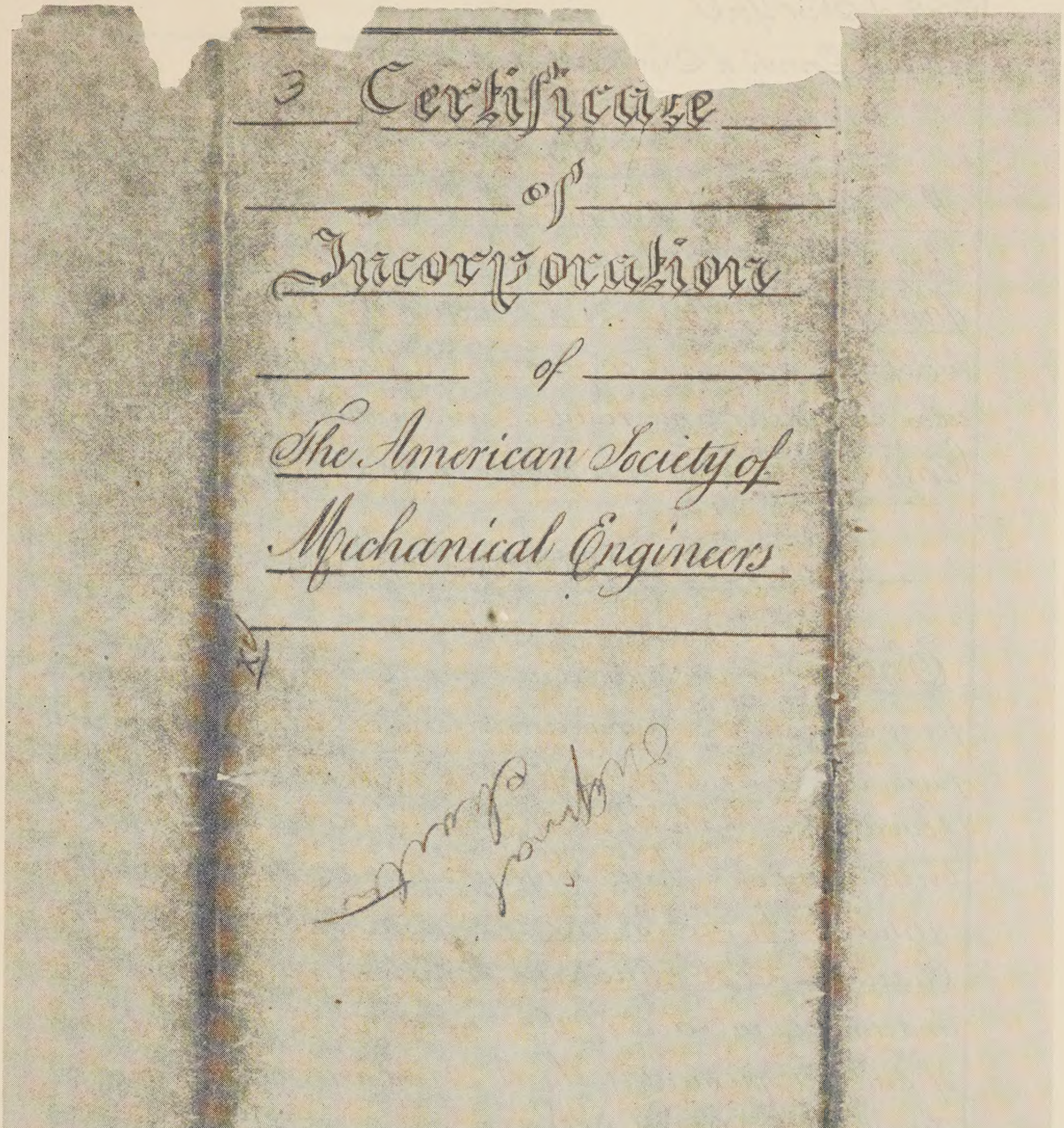
Certificate
of
Incorporation
of
The American Society of
Mechanical Engineers

FACSIMILE OF COVER OF CHARTER GRANTED THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS DECEMBER 24, 1881
The remainder of the document appears in the following pages.)

State of New York
City and County of New York } ss

We George H. Babcock William P. Trowbridge, Lycurgus B. Moore, Thomas Whiteside Rae, Alfred R. Wolff, D. S. Hines, Charles E. Emery James C. Bayles and Frederick R. Hutton all of whom are of full age, and citizens of the United States; and also citizens of and resident within this State, Except George H. Babcock who resides in the State of New Jersey:

Do Hereby Certify

That we desire to associate ourselves together, and form a Society or association for Scientific purposes, pursuant to and in conformity with an act of the Legislature of the State of New York passed on the twelfth day of April Eighteen hundred and forty-eight, entitled "An Act for the incorporation of Benevolent, Charitable, Scientific and Missionary Societies," and the several acts of the Legislature amendatory thereof and supplemental thereto, and in accordance therewith do hereby declare.

First

The name or title by which such Society shall be known in law shall be — The American Society of Mechanical Engineers.

Second.

The particular business and objects of such Society are, to promote the Arts and Sciences connected with Engineering and Mechanical Construction for Scientific purposes, and to that end to meet and associate together to read and discuss professional papers, and to circulate by means of publications among its members, the information thus obtained, and for the purpose of maintaining a library

Third.

The number of Trustees, directors or managers to manage the same shall be Eighteen; and the names of the trustees, directors or managers of such Society for the first year of its existence are, Robert H. Thurston, William H. Shock, Alexander L. Holley, Theodore N. Ely, William P. Trowbridge, Erasmus D. Leavitt Jr, Charles E. Emery, Washington Jones, William B. Cogswell, Charles B. Richards, S. B. Whiting, J. F. Holloway, George Fisher, Allan Stirling, George H. Babcock, S. W. Robinson, Charles W. Copeland and Thomas Whiteside Rae.

Fourth

The business of the said Society or Association shall be carried on in the City and County of New York, and the principal office of such Society or association shall be located in said City and County of New York.

Witness our hands and seals this twenty third day of December A D 1881

In the presence of
Wm. L. Snyder

Geo. H. Babcock
W. P. Trowbridge
Lycurgus B. Moore
Thomas Whiteside Rae
Alfred R. Wolff.
D. T. Ties
Chas E. Emery
F. R. Hutton
J. C. Bayles.

State of New York, City and County of New York. S.S.

On this 23^d day of December in the year of our Lord one thousand eight hundred and eighty one, before me personally came George H. Babcock, William P. Trowbridge, Lycurgus B. Moore, Thomas Whiteside Rae, Alfred R. Wolff, Charles E. Emery, James C. Bayles and F. R. Hutton, to me severally known to be the individuals described in and who executed the foregoing instrument and severally acknowledged that they executed the same.

Wm. L. Snyder
Notary Public
N. Y. Co.

State of New York
City and County of New York } ss.

On the 24th. day of December in the year of our Lord one thousand Eight hundred and Eightyone, before me personally came D. S. Hines

to me known to be the individual described in and who executed the foregoing instrument, and ~~severally~~ acknowledged that ~~they~~ he executed the same

Wm L. Snyder
Notary Public
N. Y. Co.

Endorsement

State of New York
City and County of New York } ss

I Abraham R. Lawrence one of the Justices of the Supreme Court of the first Judicial District of the State of New York, in which the place of business or principal office of the association or Society hereinafter mentioned shall be located do hereby certify that I have examined the Certificate of Incorporation of the Association or Society designated as "The American Society of Mechanical Engineers," and the right to establish or organize the same, under the

name and for the purposes therein mentioned, pursuant to, and in conformity with an act of the Legislature of the State of New York passed on the Twelfth day of April 1848 entitled "An act for the Incorporation of Benevolent, Charitable Scientific and Missionary Societies" and the several acts of the said Legislature amendatory thereof, and the same meets my approbation and approval, and in accordance therewith I make this endorsement. ——
Dated New York December 24 1881.

Abm R Lawrence
Justice of the
Supreme Court

Constitution, By-Laws, and Rules of The American Society of Mechanical Engineers

As Amended at Semi-Annual Meeting, Detroit, Mich., May 17, 1937

Article C1, Name and Government

Sec. 1. The name of this Society is The American Society of Mechanical Engineers.

Sec. 2. The Society is a corporation, organized April 7, 1880, and chartered under the laws of the State of New York, December 24, 1881. A supplemental charter* was issued on October 17, 1907, when the Society was consolidated with the Mechanical Engineers' Library Association.

The principal offices of the Society shall be in the City of New York.

Sec. 3. The Society shall be governed by this Constitution, the By-Laws, and the Rules.

Article B1, Government

Par. 1 Every question which shall come before a meeting of the Society or of the Council or of a committee, shall be decided by a majority of the votes cast, unless otherwise provided in the Constitution, the By-Laws and the Rules, or by the laws of the State of New York.

Par. 2 The Rules contained in "Robert's Rules of Order Revised" shall govern the Society in all cases to which they are applicable, when not inconsistent with the By-Laws or the Rules of this Society.

Article C2, Objects

Sec. 1. The objects of this Society are to promote the art and science of mechanical engineering and the allied arts and sciences; to encourage original research; to foster engineering education; to advance the standards of engineering; to promote the intercourse of engineers among themselves and with allied technologists; and severally and in cooperation with other engineering and technical societies to broaden the usefulness of the engineering profession.

Sec. 2. The Society may approve or adopt any report, standard, code, formula, or recommended practice, but shall forbid and oppose the use of its name, emblem, or initials in any commercial work or business, except to indicate conformity with its standards or recommended practices.

Article B2, Purposes

Par. 1 The objectives of the Society shall be accomplished by:

A Advancing the theory and practice of engineering and the allied arts and sciences by:

(*a*) Encouraging engineering research, tests, and other original work.

(*b*) Encouraging the preparation of original papers on engineering topics.

(*c*) Holding meetings for the presentation and discussion of original papers and participating in international engineering congresses.

(*d*) Publishing papers and reports and disseminating knowledge and experience of value to engineers.

(*e*) Developing and promulgating standards, codes, formulas, and recommended practices.

(*f*) Offering awards and other honors to encourage contributions to engineering; conferring awards and other honors in recognition of meritorious contributions to engineering.

(*g*) Furthering the purposes of the Engineering Societies' Library, of which the Library of this Society forms a part.

(*h*) Encouraging intercourse among engineers for the mutual exchange of knowledge and experience.

B Enhancing the status of the engineer by:

(*a*) Maintaining high technical and cultural standards for entrance to the Society.

(*b*) Cooperating with educational institutions in the maintenance of high standards of engineering education.

(*c*) Requiring a high standard of ethical practice by members of the Society.

(*d*) Aiding in the adoption of a high standard of attainment for the granting of the legal right to practice professional engineering.

(*e*) Fostering among engineering students the study of philosophy and history, tradition and achievement, duties, and social functions of the engineering profession.

(*f*) Encouraging the personal and professional development of young engineers.

(*g*) Supporting activities looking to the increased employment of engineers and seeking new opportunities for engineering service.

C Increasing the usefulness of the organized engineering profession by:

(*a*) Cooperating with other engineering and technical societies.

(*b*) Encouraging a high standard of citizenship among engineers.

(*c*) Encouraging engineers to participate in public affairs.

(*d*) Cooperating with governmental agencies in engineering matters.

(*e*) Publicity for the engineering profession through the achievements of engineers.

Par. 2 The closest possible cooperation with universities and technical schools qualified and equipped to assist in the development and conduct of special research work is favored and is strongly urged.

Par. 3 Cooperative, not competitive, methods should be worked out with existing research laboratories and activities in other organizations. Such cooperation could take the form of publication of papers and groups of papers where a definite industry desires to bring to the attention of engineers for the development of the industry, any problem or special research, without commercial bias.

Par. 4 Each suggested research must be presented, on its individual merit, for approval by the Council, which will in turn refer the matter to the appropriate authority or committee.

Par. 5 Specific requests to the Council for solicitation of funds are to be accompanied with full details for proposed scope, method of solicitation, and budget.

Par. 6 No contributor is to be specially favored on account of any contribution for a research in which he is interested and a contribution can be received only on the basis of general benefit to the industry.

Article R2, Affiliated Organizations

Rule 1 The Council may approve the affiliation with the Society of any engineering society or legally organized group of engineers whose objects are in accord with the traditions, precedents, and objects of this Society.

Rule 2 The term "Affiliated with The American Society of Mechanical Engineers" shall be used by any society or by individual members of it only while the respective governing boards of both socities continue the affiliation.

Rule 3 Affiliation with this Society of any other organization

* The Supplemental Charter of October 17, 1907, also provided that the number of Directors shall be twenty-two (22).

shall in no wise be interpreted as interfering with the independence, autonomy, and self-control of that organization under its own constitution or by-laws.

RULE 4 The Society shall not be responsible for any act of any affiliated society.

RULE 5 Affiliation with this Society of any other organization may be terminated by the governing board of either giving sixty (60) days' written notice to the governing board of the other.

Article C3, Membership

Sec. 1. The corporate membership shall consist of Fellows, Members, and Junior Members. In addition there shall be Honorary Members, Associates, and Student-members. The corporate membership shall also include the members of the present Associate-Member grade until this grade is automatically eliminated.*

Sec. 2. The rights and privileges of every member shall be personal to himself and shall not be transferable except that each corporate member shall be entitled to vote on any question before the Society either in person or by a proxy given to a corporate member.

Sec. 3. Every person admitted to membership shall be subject to the Constitution of the Society, and to any amendments that may be made from time to time.

ARTICLE B3, MEMBERSHIP

PAR. 1 The Council shall have power by resolution from time to time, to fix the number of Honorary Members.

PAR. 2 A proxy may be given to a member entitled to vote, but shall not be valid for more than six (6) months. Such proxy shall be signed, with an attesting witness, by the member giving it and shall be submitted to the Secretary for verification of the right of the member to vote at the meeting at which the proxy is to be used.

PAR. 3 Proffered resignations shall be presented to the Council for action, and shall be accepted if the requirements have been met. Each resignation presented to the Council after the fiscal year has commenced (October first) must be accompanied by a statement from the Secretary that the member has paid his dues up to and including the expired portion of the current fiscal year, unless such resignation is presented by January first, when no payment of current dues shall be required

ARTICLE R3, MEMBERSHIP

RULE 1 Each member shall be entitled to a certificate of membership, signed by the President and the Secretary of the Society; it shall remain the property of the Society and be returned on demand. Each member requesting a certificate shall pay the cost of engrossing.

RULE 2 Abbreviations of the titles to be used by members are as follows:

Honorary Member	Hon. Mem. A.S.M.E.
Fellow	Fellow A.S.M.E.
Member	Mem. A.S.M.E.
Associate	Assoc. A.S.M.E.
Junior Member	Jun. A.S.M.E.
Student-member	Student A.S.M.E.

RULE 3 Each member shall be entitled to wear the emblem approved by the Council for his grade of membership.

RULE 4 Each member desiring to resign shall deposit with the Secretary any badge and certificate of membership in his possession, and upon acceptance of his resignation the Secretary shall make him the stipulated refund for his badge.

Article C4, Qualifications for Admission

Sec. 1. Members of all grades shall be elected by the Council.

Sec. 2. An Honorary Member shall be a person of acknowledged professional eminence.

Sec. 3. A Fellow shall be an engineer who shall have distinct engineering attainments, twenty-five (25) years of active practice in the profession of engineering or teaching of engineering in a school of accepted standing, and shall have been thirteen (13) years in the grade of Member, or Associate-Member, or ten (10) years in the former Member grade.* Graduation from an engineering school of accepted standing shall be considered equivalent to four (4) years of active practice.

Sec. 4. An engineer lacking the qualifications of Section 3 who has distinguished engineering or scientific attainments may be elected a Fellow by unanimous vote of Council members voting.

Sec. 5. A Member shall be an engineer or teacher of engineering who shall have reached the age of thirty (30) years and who shall have had nine (9) years active practice in the profession of engineering or teaching, three (3) years of which shall have been in a position of responsible charge of important work and who is qualified to design as well as direct such work. Graduation from a school of engineering of accepted standing shall be considered equivalent to four (4) years of active practice.

Sec. 5(A). All present Associate-Members as of the date this Section is declared in effect and who shall have reached the age of thirty (30) years shall automatically be transferred to the grade of Member without fee or application, and those Associate-Members under thirty (30) years of age shall be transferred similarly as they reach the age of thirty (30) years.*

Sec. 6. An Associate need not be an engineer but must have a record of recognized leadership in some profession, or branch of industry, or science relating to engineering, and shall be qualified to cooperate with engineers in the practice of their profession and he must be at least thirty (30) years of age.

Sec. 7. A Junior Member shall be a graduate of a school of engineering of accepted standing or one who has equivalent attainments.

Sec. 8. A Student-member shall be a student regularly enrolled and pursuing an approved engineering curriculum in a school having a Student Branch of this Society.

ARTICLE B4, QUALIFICATIONS FOR ADMISSION

PAR. 1 A candidate for admission to the Society in any grade, except Honorary Membership, or a member desiring to change his grade, shall make application to the Council on an approved form.

PAR. 2 Fifteen (15) affirmative votes of the Council shall be required for the election of a candidate for any grade except Honorary Membership. Two (2) negative votes shall defeat an election.

PAR. 3 Each approved candidate shall be assigned by the Council to the grade of membership to which, in its judgment, his qualifications entitle him.

PAR. 4 Nomination for Honorary Membership may be made to the Council by at least twenty-five (25) members of the Society, who shall in all cases state in writing the grounds upon which the nomination is made.

PAR. 5 Election to Honorary Membership shall be by letter-ballot of the Council. Ballots shall be mailed by the Secretary to each member of the Council at least sixty (60) days in advance of the date set for the closure of such election. One (1) negative vote shall defeat an election to Honorary Membership.

PAR. 6 All matters relating to admissions to and promotions in membership shall be in charge of the Standing Committee on Admissions, under the direction of the Council.

PAR. 7 A Student-member may participate in all the activities of the Society but shall not be permitted to vote or hold an elective office except in the Student Branch located at the college of which he is a student.

* Underlined subject matter is to be eliminated by act of Council without ballot of membership when the transition to the new scheme of membership grades is completed.

* Underlined subject matter is to be eliminated by act of Council without ballot of membership, when the transition to the new scheme of membership grades is completed.

Par. 8 A Student-member shall not remain in this grade beyond the end of the Society's fiscal year in which he terminates his enrollment as a student.

Article R4, Qualifications for Admission

Rule 1 A candidate for admission to the Society as a Fellow, a Member, or an Associate, should refer to at least five (5) members who have personal knowledge of his qualifications and the grade of reference shall be as follows:

For Fellow, at least one (1) Fellow and the remainder Members
For Member, at least five (5) Fellows or Members
For Associate, at least three (3) Fellows or Members and the remainder Associates.

Rule 2 A candidate for admission to the Society as a Junior Member should refer to at least three (3) members who have personal knowledge of his qualifications, at least one of whom shall be a Fellow, Member, or Associate.

Rule 3 A candidate for admission to the Society as a Student-member must be endorsed by the Honorary Chairman in office at the Student Branch located at the college where he is a student.

Rule 4 An application for membership from a candidate who may not be able to give the necessary number of references may be recommended to the Council for ballot after sufficient evidence has been secured to show that the candidate is worthy of admission to membership. Such candidates may refer to officers or voting members of other societies of like standing.

Rule 5 An application may be referred by the Committee on Admissions to the executive committee of the Local Section to which the applicant would be logically attached, for information and comment by such local committee. If, after a period of twenty (20) days, no comment is received from the local committee, the Committee on Admissions will proceed with the consideration of the application.

Rule 6 The references for each candidate shall be requested to make such confidential communications to the Committee on Admissions as will enable it to arrive at a proper estimate of the eligibility of the candidate.

Rule 7 The Committee on Admissions shall report to each session of the Council the names of all candidates together with the recommendation on each. The Committee on Admissions shall meet monthly to receive and scrutinize applications, and shall seek further information as to the qualification of a candidate whose evidence of eligibility is not clear to them.

Rule 8 All confidential correspondence in relation to each candidate shall be destroyed by the administrative officer in charge of membership admissions within a reasonable period after acceptance of election by payment of the initiation or promotion fees and dues.

Rule 9 The Secretary shall mail to each member of the Council a ballot of the names and respective grades of the candidates for membership approved by the Committee on Admissions after having been duly posted in the publications of the Society. The voter shall prepare his ballot by crossing out the name of any candidate rejected by him, and shall enclose the ballot in an envelope and seal it. He shall enclose this envelope in a second envelope and sign it for identification. A ballot without the autographic endorsement of the voter on the outer envelope is defective and shall be rejected.

Rule 10 The Secretary shall count the ballots cast by the Council for election of new members, notify the applicants of their election, and regularly report the results of the ballot at the Council meeting next following each election. The names of applicants who are not elected shall neither be announced nor recorded.

Article C5, Fees and Dues

Sec. 1. Initiation Fees:

Honorary Member	**None**
Fellow	**$30**
Member	**25**
Associate	**25**
Junior Member	**10**
Student-Member	**None**

Promotion Fees:

From Member to Fellow	**5**
From Junior Member to Member or Associate	**10**
(Except that an applicant under the age of 33 who has been a Junior Member in good standing for five consecutive years may be promoted without fee.)	
From Student-member to Junior Member	**None**

Sec. 2. The annual dues for membership in each grade shall be:

Fellow	**$25**
Member	**20**
Associate	**20**
Associate-Member	**20***
Junior Member until reaching the age of 30	**10**
Junior Member between the ages of 30 and 33	**15**
Junior Member after reaching the age of 33	**20**
Student-member . . . as provided in the By-Laws	

Sec. 3. The Council may permit any Fellow, Member, or Associate to become a life member in the same grade.

Sec. 4. The Council may remit the dues of any member for any special reason.

Article B5, Fees and Dues

Par. 1 The initiation fee and that part of the annual dues from the first month following the date of election to the first day of October, shall be due and payable on the first day of the month following the date of election. Only upon the payment of this amount shall the person elected be entitled to the rights and privileges of membership in the grade to which he is assigned. If such person does not comply with this requirement within three (3) months after notice of his election, the Council may declare his election void.

Par. 2 The annual dues for each ensuing year shall be due and payable in advance on the first day of October.

Par. 3 A bill for annual dues shall be mailed to each member by October first of each year. Notice of arrears shall be sent thereafter, as directed by the Council.

Par. 4 At its first meeting in the calendar year the Secretary shall submit to the Council a list of members whose dues have remained unpaid for three (3) months. The Council may order the withholding of the publications for such delinquents.

Par. 5 At its first meeting after the close of the fiscal year on September thirtieth, the Secretary shall submit to the Council a list of members whose dues have remained unpaid for twelve (12) months. Such delinquents shall, in the discretion of the Council, be stricken from the roll of membership and shall cease to have any further rights as members.

Par. 6 If, in the case of non-payment of dues, the right to receive the publications of the Society or to vote be questioned, the books of the Society shall be conclusive evidence.

Par. 7 The Council may temporarily excuse from payment of annual dues any member who from ill health, advanced age or good reason assigned is unable to pay such dues; and the Council may remit the whole or part of dues in arrears, or accept in lieu thereof desirable additions to the Library, or collections.

Par. 8 The Council may restore to membership any person dropped from the rolls for non-payment of dues or otherwise, upon such conditions as it may deem best.

Par. 9 The annual dues for a Student-member shall be $3.00 for the fiscal year beginning October first. Eight issues of *Mechanical Engineering*, October to May inclusive, shall be included in the dues for a Student-member.

Par. 10 For distinguished service to the Society, the Council may confer life membership upon any Fellow or Member. Proposal for such action must be made at a regular meeting of the Council. Immediately following that meeting, the Secretary shall send to the members of the Council a letter ballot upon the proposal, this ballot to close in sixty (60) days. Fifteen (15) affirmative votes shall be required to approve and one (1) dissenting vote shall disapprove such proposal.

Par. 11 A Fellow, Member, or Associate may become a Life Fellow, Life Member, or Life Associate by paying the Society at one time the present worth of an annuity equal to that member's dues for the period for which he is required to pay dues.

Par. 12 The Council shall confer life membership upon any

* Underlined subject matter is to be eliminated by act of Council without ballot of membership when the transition to the new scheme of membership grades is completed.

Fellow, Member, or Associate of the Society who has paid dues for thirty-five (35) years, or who shall have reached the age of seventy (70) years after having paid dues for thirty (30) years (Student-membership years not included).

Article R5, Fees and Dues

Rule 1 A Student-member recommended by the Honorary Chairman of his Student Branch may be elected by Council to Junior membership, the election being subject to his graduation. The payment of dues for one year as Junior Member, at any time prior to September thirtieth following his graduation shall constitute acceptance of election and shall give him all the rights and privileges of the Junior Member grade from the date of such payment to October first of the following year.

Rule 2 Such payment may be divided, the Student-member paying only the initial quarter's dues ($2.50) at the time of election and the remaining three-quarters of the dues ($7.50) shall be due on September thirtieth.

Article C6, The Council (Directors)

Sec. 1. The affairs of the Society shall be managed by a Board of Directors, chosen from its membership and styled "The Council" which shall have full control of the activities of the Society, subject to the limitations of the Constitution and the results of letter ballots (Article B9, Par. 4 and Article B6, Par. 3).

Sec. 2. The Directors of the Society shall consist of a President, seven (7) Vice-Presidents, nine (9) Managers, and the last five (5) surviving Past-Presidents.

Sec. 3 The Directors may at any time, whenever sufficient cause shall appear to them, delegate to any corporate member of the Society the performance of any duties required by the Constitution to be performed by any Director.

Sec. 4. The Council shall meet immediately after the close of the Annual Meeting of the Society, at such other times as the Council may select, and at the call of the President. Eight members shall constitute a quorum of the Council.

Sec. 5. The Council shall present at the Annual Meeting of the Society a report verified by the President and the Treasurer or by twelve (12) members of the Council, showing the whole amount of real and personal property owned by the Society, where located, and where and how invested, and the amount and nature of the property acquired during the year immediately preceding the date of the report, and the manner of the acquisition; the amount applied, appropriated, or expended during the year immediately preceding such date, and the purpose, object, or persons to or for which such applications, appropriations, or expenditures have been made; also a report verified by the Secretary, giving the names and places of residence of the persons who have been admitted into membership in the Society during the year.

These reports shall be filed with the records of the Society, and an abstract shall be entered in the minutes of the proceedings of the Annual Meeting of the Society.

Article B6, The Council

Par. 1 The Council shall consider the failure of any incumbent, from inability or otherwise, to perform the duties of his office, and may, by a two-thirds vote, decree any elective office vacant. The Council shall thereupon appoint a member to fill the vacancy until the next election of officers, except for the office of the President, which shall be filled by the Vice-President serving his second year, who is senior by length of membership in the Society. Such appointment shall not render the appointee ineligible for election to any office.

Par. 2 An act of the Council which shall have received the expressed or implied sanction of the membership at the following meeting of the Society, shall be deemed to be an act of the Society and cannot afterward be impeached by any member.

Par. 3 The Council shall order the submission to the membership for decision by letter-ballot of any question of major importance involving a departure from usual custom. The Council shall appoint tellers to canvass such a ballot, the result of which shall be binding.

Article B6A, Standing and Special Committees

Par. 1 The Council shall at its first meeting of each year appoint from among its members an Executive Committee. Such committee shall consist of the President, two Vice-Presidents, and two Managers, with voting power; also the Chairman of the Finance Committee, the Chairman of the Committee on Professional Divisions and the Chairman of the Local Sections Committee, without voting power. During the intervals between sessions of the Council, the Executive Committee shall have and exercise all the general powers of the Council, except the power to fill vacancies in the Council, or to amend the By-Laws. The committee shall meet at the call of the President. The Secretary may take part in the deliberations of the Executive Committee, without vote. The Executive Committee shall keep minutes of its proceedings which shall be promptly reported to each member of the Council for approval.

Par. 2 Upon the recommendation of a business meeting of the Society or upon its own initiative, the Council shall have the power to appoint, as it may deem desirable, an Administrative Committee to assist in the conduct of the affairs of the Society. Any proposed expenditure of such a committee must be authorized by the Council before it is incurred.

Par. 3 Upon the recommendation of a business meeting of the Society or upon its own initiative, the Council shall have the power to appoint, as it may deem desirable, any Professional Committee to investigate and report upon a subject of engineering interest, except that the procedure of the American Standards Association shall be followed in organizing Sectional Committees. (See Paragraphs 10 and 11 of Article B6B.) Any proposed expenditure of such a committee must be authorized by the Council before it is incurred.

Par. 4 Administrative and Professional Committees shall be standing or special, as the By-Laws and Rules provide and the Council approves. The Chairmen of Standing Committees shall be entitled to a seat in the Council, but no vote. The term of office of one (1) member of each Standing Committee shall expire at the close of each Annual Meeting.

Par. 5 Each committee shall perform the duties required by the By-Laws and Rules, or assigned to it by the Council.

Par. 6 The Council may terminate membership on any committee on account of continued absence of the member, from inability or otherwise.

Par. 7 The President shall appoint a member to fill each vacancy in the Standing Committees.

Par. 8 Each committee shall at its first meeting elect a Chairman to serve for one (1) year.

Par. 9 A member of a Standing Committee whose term of office has expired, shall continue to serve until his successor has been elected or appointed.

Par. 10 On or before the fifteenth day of October of each year, each Standing Committee shall deliver to the Secretary a written report of its work for presentation to the Council. The Council may embody such report in its Annual Report.

Par. 11 On or before the fifteenth day of October of each year, each Special Committee shall deliver a written progress report to the Secretary for presentation to the Council. Upon receipt of this report, the Council may, in its discretion, continue the committee. The committee shall be discharged upon the adoption of the final report.

Par. 12 The Standing Committee on Finance shall, under the direction of the Council, have supervision of the financial affairs of the Society, including the books of account. The Committee shall consist of five (5) members of the Society, the term of one (1) member expiring at the close of each Annual Meeting, and two (2) members of the Council, the term of one (1) member expiring at the close of each Annual Meeting.

Par. 13 The Standing Committee on Meetings and Program shall, under the direction of the Council, have supervision of the Meetings of the Society, except business meetings. The Committee shall consist of five (5) members, and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 14 The Standing Committee on Publications shall, under the direction of the Council, have supervision of the publications of the Society. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 15 The Standing Committee on Admissions shall determine the eligibility of applicants for membership, and for transfer in membership grades, and shall make recommendation to the Council on each. The Committee shall consist of five (5) members and the term of one member shall expire at the close of each Annual Meeting.

Par. 16 The Standing Committee on Professional Divisions shall, under the direction of the Council, have supervision of the Professional Divisions of the Society. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 17 The Standing Committee on Local Sections shall, under the direction of the Council, have supervision of the Local Sections of the Society. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 18 The Standing Committee on Constitution and By-Laws shall, under the direction of the Council, have supervision of matters affecting the Constitution, By-Laws and Rules, and shall report on all matters in this connection referred to it by the Council. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 19 The Standing Committee, designated as the Board of Honors and Awards shall, under the direction of the Council, have supervision of the awards of the Society as detailed in the Rules or prescribed by the Council. Recommendations for representatives of joint bodies of award shall be made to the Council by this Board. The Board shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 20 The Standing Committee on Relations With Colleges shall, under the direction of the Council, have supervision of the Student Branches of the Society and of such work of the Society as aims to further the education of engineers through the colleges and schools of accepted standing. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 21 The Standing Committee on Education and Training for the Industries shall, under the direction of the Council, have supervision of such work of the Society as deals with education and training for the industries through agencies other than the colleges and engineering schools. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 22 There shall be a Standing Committee on Library, which shall represent the Society on the Library Board of the United Engineering Trustees, Inc. The number of members of this Committee and their terms of office shall be as required by the by-laws of the United Engineering Trustees, Inc.

Par. 23 The Standing Committee on Standardization shall advise the Council on the dimensional standardization work of the Society, including relations with the American Standards Association. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 24 The Standing Committee on Research shall advise the Council on the research work of the Society. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 25 The Standing Committee on Safety shall advise the Council on the activities of the Society having to do with engineering and industrial safety, except the activities of the Boiler Code Committee, for which special provision is made. This Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

Par. 26 The Special Committee on Boiler Code shall, under the direction of the Council, have supervision of all the activities of the Society in connection with the A.S.M.E. Codes for Pressure Vessels, including the interpretations of these codes. The Committee shall be appointed by the President and confirmed by the Council, and the President shall fill all vacancies in the Committee.

Par. 27 The Standing Committee on Power Test Codes shall, under the direction of the Council, have supervision of all the activities of the Society in connection with the A.S.M.E. Power Test Codes, including the interpretation of such codes. The Committee shall consist of twenty-five (25) members and the terms of five (5) members shall expire at the close of each Annual Meeting.

Par. 28 The Standing Committee on Professional Conduct shall, under the direction of the Council, have supervision of all matters relating to the Code of Ethics and its enforcement. The Committee shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting.

B6B, Society Representation

Par. 1 The Council may, in its discretion, appoint a member or members, or other person or persons, to represent it at meetings of societies of kindred aim or at public functions. Such delegates shall be designated as "Honorary Vice-Presidents," and their duties shall terminate with the occasion for which they are appointed.

Par. 2 The President, subject to the approval of the Council, may nominate or appoint a member or members, or other person or persons, to represent the Society on professional or other committees organized by other societies or by Government departments or bureaus, or otherwise.

Par. 3 The Council shall elect three (3) Trustees to serve on the Board of Trustees of the United Engineering Trustees, Inc., as required in the by-laws of that body.

Par. 4 The Council shall appoint delegates to serve on the American Engineering Council as required in the by-laws of that body. If the number of delegates required to serve is at variance with the number elected or in office, the Council of the Society is empowered to make adjustments necessary. The President of the Society in office shall be the Chairman of the delegation of this Society to the meetings of the American Engineering Council, and the Chairman of the A.S.M.E. representatives on the Executive Board.

Par. 5 The Council shall designate the Standing Committee on Library to serve as the Society's representatives on the Library Board of the United Engineering Trustees, Inc., as required in the by-laws of that body.

Par. 6 The Council shall nominate to the United Engineering Trustees, Inc., two (2) members of the Society to serve on the Engineering Foundation as required in the by-laws of that body.

Par. 7 The Council shall appoint such number of members to represent the Society on the following agencies as may be required by the by-laws of those bodies, namely: John Fritz Medal Board of Award, Washington Award Commission of the Western Society of Engineers, Gantt Medal Board of Award, Daniel Guggenheim Medal Fund, Inc., Hoover Medal Board of Award, Alfred Noble Prize.

Par. 8 The Council shall nominate three (3) members to represent the Society on the Division of Engineering of the National Research Council as required in the by-laws of that body.

Par. 9 The Council shall appoint three (3) members of the Society to represent the Society on the Engineers' Council for Professional Development.

Par. 10 The Council shall designate such number of members to represent the Society on the American Standards Association as may be required by the constitution of that body.

Par. 11 The representatives of the Society on Sectional Committees, organized under the rules of the American Standards Association, shall be appointed by the President, subject to the approval of the Council.

Article C7, Election of Directors

Sec. 1. The membership of the Society shall elect annually a Regular Nominating Committee, whose duty shall be to select candidates for the executive offices to be filled at each annual election.

Sec. 2. Other nominating committees having the same powers may be constituted by the membership of the Society.

Sec. 3. The Directors shall be elected at the Annual Meeting of the Society, on the first Tuesday in December, as provided in the Charter.

The election shall be by sealed letter-ballot of the membership.

Sec. 4. The President shall be elected for one (1) year, the Vice-Presidents for two (2) years, the Managers for three (3) years. The Council shall have power to fill vacancies in its membership by appointment until the next election, except that the office of president shall be filled by the vice-president who is senior by length of membership in the Society.

Article B7, Election of Directors

Par. 1 The Regular Nominating Committee of the Society shall consist of seven (7) members with seven (7) alternates elected at the Annual Meeting. The Chairman of the outgoing Nominating Committee shall serve as an advisory member, without vote, and the Secretary of the outgoing Committee may serve as alternate for him.

Par. 2 The members and alternates of the Regular Nominating Committee shall be elected for one (1) year, and no member or alternate shall be eligible for more than two (2) consecutive terms. Serving as an alternate shall not affect the eligibility of a member to serve on the committee for two (2) terms.

Par. 3 For the purpose of nominating members of the Regular Nominating Committee, the Council shall, on or before the first day of October of each year, associate the Local Sections into seven (7) groups, each group to be responsible for nominating one (1) member of the Committee. The Sections which will comprise these groups shall, as far as possible, be contiguous geographically to each other.

Par. 4 The names of those elected to serve on the Regular Nominating Committee shall be published by the Secretary by the first week in February of each year, accompanied by a request for suggestions for nominees.

Par. 5 A vacancy in a Regular Nominating Committee of the Society shall be filled by the alternate for that vacancy, or failing that, shall be filled by the Council.

Par. 6 A Special Nominating Committee may be organized by any group of one (1) per cent of the membership of the Society in good standing certifying to the Secretary in writing their joint intention to organize such a Committee.

Par 7 Within two weeks following the Semi-Annual Meeting, the Regular Nominating Committee shall deliver to the Secretary in writing the names of its nominees for the elective offices to be filled at the next election, together with the written consents of the nominees.

Par. 8 The names and qualifications of nominees for the various offices proposed by the Regular Nominating Committee shall be published by the Secretary immediately after the receipt of the report of the Nominating Committee.

Par. 9 Candidates for the office of President and of Vice-President shall be of the grade of Fellow or of Member of the Society Candidates for all other elective offices may be of any grade of corporate membership.

Par. 10 Names of any nominees presented by any Special Nominating Committee must be in the hands of the Secretary by the first Tuesday in August of each year, and must be accompanied by the written consent of each nominee.

Par. 11 On or before the third Thursday in August of each year, the Secretary shall mail to each member entitled to vote a ballot stating the names of the candidates for the elective offices to be filled at the next election.

Par. 12 Voting for the election of Directors shall close at the City of New York at 10 o'clock in the forenoon on the fourth Tuesday in September in each year, and the ballots shall be canvassed.

Par. 13 On or before the third Thursday in August of each year, the President shall appoint three (3) Tellers of Election of Directors, whose duty it shall be to canvass the votes cast. The term of office of the Tellers shall expire when their report of the canvass has been presented and accepted.

Par. 14 By the first day of October, the Secretary shall notify the candidates having the greatest number of votes for their respective offices.

Par. 15 The Directors shall be declared elected by the Presiding Officer at the Annual Meeting of the Society in December, and their terms of office shall begin on the adjournment of the Annual Meeting.

Par. 16 If a tie occurs in the vote for any officer, the Presiding Officer at the Annual Meeting shall cast the deciding vote.

Par. 17 In the election of the Vice-Presidents, three (3) shall be elected every other year and four (4) the alternate years to serve for two (2) years.

Par. 18 In the election of the Managers, three (3) shall be elected each year to serve for three (3) years.

Par. 19 A member in office shall not be eligible for immediate reelection to the same office at the expiration of the term for which he was elected, except the Secretary and the Treasurer.

Par. 20 Members in office shall continue in their respective offices until their successors have been elected or appointed, and have accepted their offices.

Article R7, Election of Directors

Rule 1 The Chairman of the Committee on Local Sections, or in his absence, the senior member of the Committee, shall preside at the Conference of Group Representatives at the time action is taken on the Regular Nominating Committee.

Rule 2 At the business session of the Annual Meeting of the Society, the Chairman of the Committee on Local Sections shall present names recommended by the conference for the Regular Nominating Committee

Rule 3 The names of the candidates proposed by the Regular Nominating Committee and by any other nominating committee, and the respective offices for which they are candidates, shall be printed in separate lists on the same ballot sheet, each list of candidates to be printed under the names of the members of the particular committee which proposed it.

Rule 4 Each list of names shall contain the name of only one (1) candidate for the office of President. For any other office than President, there may be more than one (1) candidate.

Rule 5 In the election of Directors, the voter shall prepare his ballot by crossing out the name of any candidate or candidates rejected by him and may write in the name of any eligible member of the Society, and shall enclose the ballot in an envelope and seal it. He shall then enclose this envelope in a second envelope marked "Ballot for Directors" and seal it, and he shall then write his name thereon for identification.

Rule 6 The Tellers shall not receive any ballot after the stated time for the closure of the voting.

Rule 7 The Secretary shall certify to the competency and signature of all voters.

Rule 8 The Tellers shall open and destroy the outer envelopes and then open the inner envelopes and canvass the results.

Rule 9 A ballot without the autographic endorsement of the voter on the outside envelope is defective and shall be rejected by the Tellers of Election.

Rule 10 A ballot containing more names than there are offices to be filled is defective and shall be rejected by the Tellers.

Rule 11 In counting the ballots for officers, the Tellers shall consider a ballot for any officer as valid providing the intent of the voter as to that particular office is clear, even though his ballot as to candidates for another office may for any reason be invalid.

Article C8, Officers

Sec. 1. The Officers of the Society shall consist of the President, the Vice-Presidents, the Secretary, and the Treasurer.

Sec. 2. At its first meeting after the Annual Meeting of the Society the Council shall appoint members of the Society to serve as Secretary and as Treasurer for one (1) year.

Sec. 3. Any vacancy in the office of Secretary or Treasurer shall be filled by appointment by the Council.

B8 Officers

Par. 1 The Officers shall perform the duties regularly or customarily attaching to their offices under the laws of the State of New York, and such other duties as may be required of them by the Council or the By-Laws.

Par. 2 In the absence of the President his duties shall be per-

formed by the Vice-President then present, who is serving his second year and is senior by length of membership in the Society, or in his absence or any other disability, by any other member of the Council designated by the Council.

PAR. 3 The Secretary and the Treasurer shall take part in the deliberations of the Council but shall have no vote therein.

PAR. 4 The Treasurer shall be the legal custodian of all funds of the Society. The investment of all trust funds and of other permanent or temporary investment of funds shall be made by the Treasurer with the approval of the Finance Committee and the Council.

PAR. 5 In the absence of the Treasurer his duties shall be performed by any other officer of the Society designated by the Council.

PAR. 6 The Secretary of the Society shall be the Secretary of the Council and of each of the committees.

PAR. 7 The Secretary shall receive a salary which shall be fixed by the Council.

PAR. 8 Any officer may be subject to removal for cause by a vote of fifteen (15) members of the Council at any time, after one (1) month's written notice has been given him to show cause why he should not be removed, and after he has been heard in his own defense, if he so desires.

R8, SECRETARY'S OFFICE

RULE 1 The office of the Secretary shall be open on business days from 9 a.m. to 5 p.m.; on Saturdays from 9 a.m. to 1 p.m.

RULE 2 The Secretary shall establish and enforce rules for the conduct of the business of his office.

RULE 3 The Secretary shall have charge of the rooms of the Society and furnishings, the historical relics and objects of art, and shall make suitable recommendations to the Council for their care and use.

Article C9, Meetings of the Society

Sec. 1. The Annual Meeting of the Society shall be held at such time and place as the Council shall appoint, provided it begins in the City of New York and continues there during the annual election of directors, held on the first Tuesday in December.

Sec. 2. The Semi-Annual Meeting of the Society shall be held at such time and place as the Council shall appoint.

Sec. 3. A special business meeting of the Society may be called at any time and place at the discretion of the Council, or shall be called by the Secretary upon the written request of at least one (1) per cent of the membership.

The call for the meeting shall be issued at least thirty (30) days prior to the date set for it, and shall state the business to be considered. No other business shall be transacted at the meeting.

Sec. 4. There shall be a business meeting of the Society during the Annual Meeting and during the Semi-Annual Meeting. At business meetings fifty (50) corporate members shall constitute a quorum.

Sec. 5. An action of a business meeting of the Society shall be deemed an action of the Society as a whole. Any expenditure required by such action is subject to approval and authorization by the Council.

Sec. 6. A General Meeting of the Society, primarily for the presentation and discussion of technical papers, may be held at such time and place as the Council shall appoint.

ARTICLE B9, MEETINGS OF THE SOCIETY

PAR. 1 An Annual Meeting may be adjourned to any other city than the City of New York upon the recommendation of the Committee on Meetings and Program, and upon authorization by the Council.

PAR. 2 A Semi-Annual Meeting shall be held upon the recommendation of the Committee on Local Sections, confirmed by the Committee on Meetings and Program, and authorized by the Council at its regular meeting at the previous Semi-Annual Meeting.

PAR. 3 A General Meeting shall be held upon the recommendation of the Committee on Local Sections, confirmed by the Committee on Meetings and Program, and authorized by the Council.

PAR. 4 Any business meeting of the Society at which a quorum is present may order the submission of any question to the membership for letter-ballot, and the result of the ballot shall be binding.

PAR. 5 Announcements of all Meetings of the Society shall be made in the publications. A notice of each meeting shall be given by the Secretary to each member not less than thirty (30) days before the date of that meeting.

PAR. 6 All Meetings of the Society, except business meetings, shall be in charge of the Committee on Meetings and Program, under the direction of the Council.

ARTICLE R9, MEETINGS OF THE SOCIETY

RULE 1 Subject to the approval of the Committee on Meetings and Program, any Local Sections participating in the conduct of a Semi-Annual or General Meeting shall appoint the necessary special local committees which shall function under the direction of the Committee on Meetings and Program.

Article C10, Professional Divisions

Sec. 1. The Council may authorize the organization of Professional Divisions composed of members of any or all grades which shall operate under the provisions of the Constitution, By-Laws, and Rules.

ARTICLE B10, PROFESSIONAL DIVISIONS

PAR. 1 The object of each Professional Division shall be to provide, through an organization of members of any or all grades particularly interested in a branch of engineering included in the scope of the Society's activities, means for promoting the arts and sciences of that branch.

PAR. 2 A Professional Division of the Society may be organized upon acceptance by the Council of the written request of a satisfactory number of members. Such a Division shall be designated as the .... Division of The American Society of Mechanical Engineers.

PAR. 3 The provisions of the Constitution, By-Laws, and Rules of the Society shall cover the procedure of all Professional Divisions, but no action or obligation of a Division shall be considered an action or obligation of the Society as a whole. This By-Law shall be imprinted on any publication issued by a Division.

PAR. 4 For the convenient conduct of its affairs, each Professional Division shall organize an executive committee. The executive committee shall elect its Chairman each year, and upon confirmation by the Council, he shall serve as Chairman of the Division.

PAR. 5 The function of the Standing Committee on Professional Divisions, under the direction of the Council, shall be to organize, foster, and coordinate Professional Divisions and their activities.

PROFESSIONAL GROUPS

PAR. 6 In case the number of members interested in a particular branch of the Society's work is not large enough to warrant the formation of a full Professional Division under the provisions of the By-Laws, the Council may authorize the formation of a Professional Group, and will itself appoint an executive committee to organize such a Group, and will designate the Chairman of the Committee. When a sufficient number of members become attached to this Group, it may petition for reorganization into a Professional Division.

ARTICLE R10, PROFESSIONAL DIVISIONS

RULE 1 When a number of members of the Society interested in a particular branch of the work of the Society favor the formation of a Professional Division for that branch, they may draw up a petition for the establishment of such a Division. Each such petition shall be sent to the Standing Committee on Professional Divisions for presentation to the Council with its recommendation. Upon approval of the petition by the Council, the Chairman of the Standing Committee on Professional Divisions shall appoint a temporary Chairman of the new Division.

RULE 2 The executive committee of each Professional Division shall consist of five (5) members and the term of one (1) member shall expire at the close of each Annual Meeting. The executive committee and such officers as the Division may require shall be selected from the membership of the Society. Other committees, advisors, and associates of the Division shall be appointed by the executive committee as required for a term not exceeding one (1) year.

RULE 3 Upon the organization of a Professional Division the ini-

tial selection of the executive committee shall be made by the President upon the nomination of the Standing Committee on Professional Divisions which will state the length of term of each appointee.

Rule 4 During the month of October of each year the executive committee of each Division will nominate to the President through the Standing Committee on Professional Divisions one or more individuals from whom the President shall appoint the member of the executive committee.

Rule 5 The executive committee of each Professional Division shall elect its own officers. No one shall be eligible for chairmanship until he has been a member of this committee for one year, except in the selection of the executive committee for a newly formed Division.

Rule 6 In case of resignation or decease, vacancies shall be filled by appointment of the executive committee subject to the approval of the President of the Society.

Rule 7 The executive committee may, subject to the approval of the Secretary of the Society, appoint or elect a secretary of the Division, who shall report the proceedings of that Division to the Secretary of the Society for notice in the publications. He shall perform the duties of secretary of the Division, and such other duties as may be prescribed by the executive committee.

Rule 8 Any expenditure for the purpose of a Division chargeable to the Society must be authorized by the Secretary of the Society before it is incurred, and must be provided for in the annual budget approved by the Council. Any liability otherwise incurred shall not be binding on the Society, and must be met by the Division itself.

Rule 9 Notice of all Professional Division meetings shall be given in writing to the Secretary of the Society and to the Chairman of the Standing Committee on Professional Divisions at least six (6) weeks in advance of the date set for such meetings.

Professional Groups

Rule 10 The functions and responsibilities of a Professional Group shall be the same as those of a Professional Division, except that the Chairman of the executive committee, although having a seat in the conferences of the Chairmen of the Professional Divisions, shall have no vote.

Rule 11 The activities of a Professional Group shall be subject to the jurisdiction of the Standing Committee on Professional Divisions.

Rule 12 The Council reserves the right to disband any Professional Group on sixty (60) days' notice.

Article C11, Local Sections

Sec. 1. The Council may authorize the organization of Local Sections composed of members of any or all grades, which shall operate under the provisions of the Constitution, By-Laws, and Rules.

Sec. 2. The Local Sections shall be associated into geographical groups with annual meetings at which each Local Section shall be entitled to representation. Each geographical group shall be entitled to representation in a subsequent conference of group representatives at the Annual Meeting of the Society.

Article B11, Local Sections

Par. 1 The object of a Local Section of the Society shall be to provide means for promoting the work of the Society by a local organization of members who are resident within a given territory.

Par. 2 A Local Section shall consist of members of any or all grades and may include other persons.

Par. 3 A Local Section of the Society may be organized upon acceptance by the Council of the written request of a satisfactory number of members. Such a Section shall be designated as the .... Section of The American Society of Mechanical Engineers.

Par. 4 The provisions of the Constitution, By-Laws, and Rules of the Society shall cover the procedure of all Local Sections, but no action or obligation of a Section shall be considered an action or obligation of the Society as a whole. This By-Law shall be imprinted on any publication issued by the Section.

Par. 5 For the convenient conduct of its affairs, each Section shall organize an executive committee.

Par. 6 The affairs of the Local Sections shall be in general charge of the Standing Committee on Local Sections, under the direction of the Council.

Par. 7 The Council of the Society, on sixty (60) days' notice, may suspend or disband any Local Section.

Par. 8 There shall be a group meeting of Local Section delegates of each group preferably between October fifteenth and November fifteenth of each year at some central point within the geographical limits of the group.

Par. 9 Each Local Section shall be entitled to one voting delegate in the group meeting of Local Section delegates. In addition to such delegates the group representative serving his second year shall serve as chairman of the group meeting, but shall vote only in case of a tie.

Par. 10 At such group meeting of Local Section delegates, one representative shall be elected to represent the group for two (2) years at the conference of group representatives.

Par. 11 There shall be a conference of group representatives each year at the place and at the time of the Annual Meeting of the Society. There shall be fourteen (14) representatives to such annual conference, two (2) from each of the seven (7) groups, which groups shall conform geographically to those provided for in Article B7, Par. 3.

Article R11, Local Sections

Rule 1 When a number of members of the Society in any territory within the limits of North America, Hawaii, Puerto Rico, and Cuba favor the formation of a Local Section in that territory, a preliminary meeting shall be called and notice sent to the entire membership of the Society residing in that territory. At this meeting a petition for the formation of a Local Section, containing suggestions as to the territory to be included in the Section, may be presented, and, if adopted, shall be sent to the Standing Committee on Local Sections for recommendation to the Council.

Rule 2 Upon the approval by the Council of the petition, a meeting of the signers shall be held for the selection of a temporary executive committee of at least five (5) members. This committee shall have charge of, and be responsible for, the proceedings of the Local Section until the next election of officers.

Rule 3 The executive committee of a Local Section shall consist of a chairman, a secretary, and such other officers as may be found desirable. Such officers shall be elected by ballot of the members of the Society constituting the Section. The committee shall be elected before the first day of June each year and shall take office on the first day of July.

Rule 4 A member of the Society shall be entitled to vote or to hold office in not more than one (1) Local Section at a time.

Rule 5 The chairman of each Local Section shall have the privilege of attending all meetings of the Standing Committee on Local Sections.

Rule 6 The secretary of each Local Section shall report the proceedings of that Section to the Secretary of the Society for notice in the publications. He shall discharge the duties of secretary of the Section, and such other responsibilities as may be prescribed by the executive committee.

Rule 7 Any expenditure chargeable to the Society for the purpose of any Local Section must be provided for in the annual budget approved by the Council. No liability otherwise incurred shall be binding upon the Society.

Rule 8 Each Local Section shall use only such uniform stationery as is supplied by the Secretary of the Society.

Rule 9 For the convenient cooperation between the Local Sections and the Professional Divisions, each Local Section may appoint an individual or a committee to act as a correspondent with each Professional Division, with duties that will comprise generally the arranging with the Professional Division for the presentation of papers, holding of meetings, etc., within that particular Local Section, and as far as possible, to act as a means of furnishing information, secured within the Local Section, which might prove of interest to the Division.

Rule 10 A Local Section may affiliate with existing local engineering organizations, or form jointly with them new local engineering organizations, but the plan of such affiliation or organization, and the obligations assumed by the Local Section and the Society thereby, shall first be approved by the Council on recommendation of the Committee on Local Sections. Any expenditures incurred in such an affiliation shall be binding only on the Section and not on the Society as a whole.

Rule 11 A Local Section may arrange to hold joint meetings with other engineering organizations and may invite members of such organizations to attend its meetings, but all expenses incurred shall be binding only on the Section and not on the Society as a whole.

Rule 12 Each Local Section may adopt its own by-laws, for the conduct of its affairs, provided such are in harmony with the Constitution, By-Laws, and Rules of the Society, and provided also every publication of such by-laws be prefaced with a copy of this Rule.

Rule 13 Groups of members residing outside the limits of North America, Hawaii, Puerto Rico, and Cuba may engage in group activities with local members of the A.S.C.E., A.I.M.E., and A.I.E.E., in which case the Council may grant them nominal financial support, provided such group action is not in conflict with the policies and activities of any established national engineering societies in such foreign countries, and that such groups cooperate as permitted with such foreign societies.

Article C12, Student Branches

Sec. 1. The Council may authorize the organization of Student Branches which shall operate under the provisions of the Constitution, By-Laws, and Rules.

ARTICLE B12, STUDENT BRANCHES

PAR. 1 A Student Branch may be organized upon acceptance by the Council of the written request of at least fifteen (15) senior and junior students in any engineering school of accepted standing. Such a Branch shall be designated as the .... Student Branch of The American Society of Mechanical Engineers.

PAR. 2 The provisions of the Constitution, By-Laws, and Rules of the Society shall cover the procedure of all Student Branches, but no action or obligation of a Student Branch shall be considered an action or obligation of the Society as a whole. This By-Law shall be imprinted on any publication issued by the Student Branch.

PAR. 3 The function of the Standing Committee on Relations With Colleges under the direction of the Council shall be to organize, foster, and govern Student Branches and their activities.

PAR. 4 Annual regional conferences of delegates from Student Branches shall be held at the discretion of the Committee on Relations with Colleges.

ARTICLE R12, STUDENT BRANCHES

RULE 1 Upon the recommendation of a Student Branch, the President of the Society shall designate a corporate member of the Society as Honorary Chairman for one (1) year, to be a member ex officio of the governing body of the Student Branch.

RULE 2 Annually, each Student Branch shall select officers including a chairman and a governing body of at least three (3) Student-members.

Article C13, Publications and Papers

Sec. 1. The papers and publications of the Society shall be issued in such manner as the Council may direct.

ARTICLE B13, PUBLICATIONS AND PAPERS

PAR. 1 All publications of the Society shall be in charge of the Standing Committee on Publications, under the direction of the Council.

PAR. 2 The publications of the Society shall consist of (*a*) the Transactions of the Society; (*b*) Mechanical Engineering; and (*c*) such other publications as the Council may direct.

PAR. 3 The policy of the Society shall be to give papers read before it the widest publicity.

PAR. 4 The Society shall not be responsible for statements or opinions advanced in papers or in discussion at meetings of the Society or of its Divisions or Sections, or printed in its publications.

PAR. 5 The Society reserves the right to copyright, at the discretion of the Council, any of its papers, discussions, reports, or publications.

Article C14, Funds

Sec. 1. The deposit, investment, and disbursement of all funds shall be subject to the direction of the Council.

ARTICLE B14, FUNDS

RECEIPTS

PAR. 1 All funds shall be paid in to the Secretary, who shall enter them in the books of the Society, and deposit them to the account of the Treasurer in a bank designated by the Council.

PAR. 2 All bills against members and others shall be made and collected by the Secretary.

PAR. 3 Income from initiation fees shall not be used for current expenses. At the close of each fiscal year, unless the Council then orders otherwise, this income shall be added to surplus.

PAR. 4 Funds may be solicited from sources outside of the Society for the conduct of research. All contributions to the Society for any specific purpose shall be disbursed under the direction of the Council.

PAR. 5 All gifts or bequests not designated for a specific purpose shall be invested and only the income shall be used.

PAR. 6 All gifts or bequests to the Society designated by the donors for a specific purpose, and all moneys permanently set aside by the Council for specific purposes, shall be invested and either the capital or income as so designated shall be used for that specific purpose for which it was designated.

PAR. 7 Except where otherwise definitely specified in a gift or bequest, the Secretary of the Society shall at the close of each fiscal year compute the interest and return received for the year on the Society's invested funds. The Secretary shall determine an average rate of income and shall recommend an apportionment of such return to each of the several funds for which investment is made. Upon approval and order of the Council these apportioned returns shall be duly entered in the books of account of the Society as the income for the year on the various funds.

PAR. 8 At the discretion of the Council income from any fund may be allowed to accumulate for expenditure in any subsequent year, or the income may be added to the original fund and invested with it. But in no case shall money be expended from such specially designated funds, either from capital or from income duly apportioned as detailed in paragraph 7, for the current expenses of the Society.

PAR. 9 Upon the maturity of any permanent investment the Treasurer shall reinvest such funds subject to approval by the Finance Committee and the Council.

PAR. 10 The securities of the Society, either principal or trust funds, may be sold, bought, or exchanged upon the written order of the Treasurer, the Secretary, and the Chairman of the Finance Committee, and these three signatures must appear on any order to any broker, bank, or company. If any one or two of these officers be temporarily unavailable, then an equal number of members of the Executive Committee may be substituted

EXPENDITURES

PAR. 11 All expenditures shall be made in accordance with the budget of appropriations as adopted by the Council.

PAR. 12 Any obligations which may be incurred during the fiscal year and which will require the expenditure of the Society's funds outside of appropriations made by the Council shall first be referred to the Finance Committee for report by that Committee back to the Council.

PAR. 13 The Secretary shall report to the Council each month the total obligations incurred against each appropriation, together with the amount of each appropriation which is unexpended.

PAR. 14 The annual appropriations approved by the Council, or so much thereof as may be required for the work of the Society, shall be expended by the Secretary, under direction of the committees.

PAR. 15. All bills against the Society shall be in charge of the Secretary who shall present them in proper form to the Finance Committee for audit.

PAR. 16 Funds of the Society shall be paid out by the Treasurer only upon vouchers duly signed by the Secretary and audited by the Finance Committee.

ARTICLE R14, FUNDS

RULE 1 The accounts of the Society shall be audited and approved annually by a chartered or other competent public accountant.

RULE 2 The Finance Committee shall hold monthly meetings for the auditing of bills and such other business as shall come before it.

RULE 3 Each year the Finance Committee shall present with its report a detailed estimate of the probable income and expenditures of the Society for the following twelve (12) months.

RULE 4 Any contract or other obligations to pay money in the Society's work shall be valid only when signed by the Secretary.

Article C15, Professional Practice

Sec. 1. In all professional and business relations the members of the Society shall be governed by the Code of Ethics incorporated in the By-Laws.

Sec. 2. Any member who has violated the Constitution of the Society, or who is guilty of conduct rendering him unfit to remain a member, may be expelled by the vote of fifteen (15) members of the Council, after he has been given opportunity to be heard in his own defense.

ARTICLE B15, PROFESSIONAL PRACTICE

PAR. 1 All members of the Society shall subscribe to the following Code of Ethics, as required by the Constitution:

A CODE OF ETHICS FOR ENGINEERS

That the dignity of his chosen profession may be maintained, it is the duty of every engineer

1 To carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, and devotion to high ideals of personal honor.

2 To refrain from associating himself with, or allowing the use of his name by, any enterprise of questionable character.

3 To treat as confidential his knowledge of the business affairs or technical processes of clients or employers when their interests require secrecy.

4 To inform a client or employer of any business connections, interests, or affiliations which might influence his judgment or impair the disinterested quality of his services.

5 To accept financial or other compensation for a particular service from one source only, except with the full knowledge and consent of all interested parties.

6 To advertise only in a dignified manner, to refrain from using any improper or questionable methods of soliciting professional work, and to decline to pay or to accept commissions for work secured by such improper or questionable methods.

7 To refrain from using unfair means to win professional advancement and to avoid unfairly injuring another engineer's chances to secure and hold employment.

8 To cooperate in building up the engineering profession by the interchange of general information and experience with his fellow engineers and with students of engineering and also by contributions to the work of engineering societies, schools of applied science, and the technical press.

9 To interest himself in the public welfare and to be ready to apply his special knowledge, skill, and training in the public behalf for the use and benefit of mankind.

Article R15, Professional Practice

Rule 1 The Standing Committee on Professional Conduct, having in charge all matters connected with the Code of Ethics and its enforcement, shall cooperate with similar committees of other societies.

Rule 2 The Standing Committee on Professional Conduct shall follow the procedure below in considering cases presented to it:

(*a*) Cases for consideration may be:

(1) An interpretation of the code, or
(2) Rendering an opinion on the questionable conduct of a member of the Society.

(*b*) Cases and complaints are to be submitted to the Committee by the Secretary of the Society.

(*c*) Before a case is submitted to the Committee, the Secretary of the Society shall ascertain whether the person against whom a complaint has been made is a member of the Society, and if possible decide whether the case is of such importance as to be passed on by the Committee, or is of such a trivial nature that it can be handled by the Secretary.

(*d*) A case may be submitted by the Secretary of the Society either through the Chairman or jointly to each member of the Committee.

(*e*) On receipt of a case the Committee shall decide whether it can best make a finding by correspondence, or by a meeting of the Committee, and whether hearings shall be given to the interested parties.

(*f*) The Committee may appoint subcommittees to consider and report on cases too remote for the main Committee to act upon.

(*g*) All correspondence from members of the Committee should pass through the office of the Chairman of the Committee and not be sent direct to the Secretary of the Society. In order to facilitate filing and preparation of reports, a letter should cover only one case or subject.

(*h*) Reports and findings on cases shall be sent by the Chairman to the Secretary of the Society for consideration by the Executive Committee or Council of the Society, which may approve the findings or take such other action as may seem desirable or necessary.

(*i*) The Committee may, if it so desires, suggest action by the Executive Committee or Council.

(*j*) The Council shall have the power on recommendation of the Committee on Professional Conduct, either (1) to censure by letter the conduct of a member who has acted contrary to the Code, if the breach is of a minor character, or (2) to cause the member's name to be stricken from the rolls of the Society, as provided in C15, Sec. 2.

Article C16, Amendments to the Constitution

Sec. 1. At any Meeting of the Society any person entitled to vote may propose in writing an amendment to this Constitution, provided that it shall bear the written indorsement of at least twenty (20) corporate members in good standing.

Such propsed amendment shall not be voted on for adoption at that meeting, but shall be open to discussion and modification, and to a vote as to whether, in its original or modified form, it shall be mailed in printed form to the members of the Society for action.

If the members present at the meeting, not less than twenty (20) voting in favor thereof, shall so decide, then the Secretary shall mail in printed form to each person entitled to vote, at least sixty (60) days previous to the next Meeting of the Society, a copy of the proposed amendment as so decided by said vote, accompanied by any comment the Council may elect to make.

A ballot shall be sent with the proposed amendment, and the voting shall be by sealed letter-ballot, closing at noon of the twentieth (20th) day preceding the meeting of the Society following the mailing.

The adoption of the amendment shall require a vote in its favor of two-thirds of the votes cast.

The Presiding Officer at the meeting of the Society following the close of the ballot shall announce the result, and if the amendment is adopted it shall thereupon take effect.

Sec. 2. Any changes in the order or numbering of paragraphs of the Constitution, By-Laws, and Rules required by an amendment shall be made under the direction of the Council.

Sec. 3. This Constitution shall supersede all previous rules of the Society, and shall go into effect upon the adjournment of the meeting of the Society at which the presiding officer announces its adoption.

Article B16, Amendments

Par. 1 At least fourteen (14) days before the closing of a ballot on an amendment to the Constitution, the President shall appoint three (3) Tellers whose duty it shall be to canvass the votes cast.

Par. 2 The Tellers shall canvass the ballots and shall certify the result to the Presiding Officer at the meeting of the Society at which the result is to be announced.

Par. 3 In the case of a tie vote on an amendment, the Presiding Officer at the Meeting of the Society shall cast the deciding vote.

Par. 4 The terms of office of the Tellers shall expire when their report of the canvass has been presented and accepted.

Par. 5 At any regular meeting, the Council may, by a two-thirds vote of its members present, adopt, or amend By-Laws in harmony with the Constitution, provided that such By-Laws or amendments shall have been submitted in writing at a previous meeting of the Council and the Secretary has mailed a copy to each member of the Council at least fifteen (15) days before the meeting at which action is to be taken. A By-Law or an amendment to a By-Law shall take effect immediately upon its adoption by the Council, and shall be published at once by the Secretary to all members of the Society.

Par. 6 At any regular meeting, by a majority vote of its members present, the Council may adopt or amend Rules in harmony with the Constitution and the By-Laws. A Rule or an amendment shall take effect immediately upon its adoption by the Council, and shall be published by the Secretary to all the members of the Society.

Article R16, Amendments

Rule 1 In voting on an amendment to the Constitution the voter shall prepare his ballot by crossing out that part of the amendment which he wishes to vote against. He shall then enclose the ballot in an envelope and seal it, and shall enclose this envelope in a second envelope marked "Ballot on Amendment" and seal it, and he shall then write his name thereon for identification.

Rule 2 The Tellers shall not receive any ballot after the stated time for the closure of the voting.

Rule 3 The Secretary shall certify to the competency and signature of all voters.

Rule 4 The Tellers shall open and destroy the outer envelopes and then open the inner envelopes and canvass the results.

Rule 5 A ballot without the autographic endorsement of the voter on the outside envelope is defective and shall be rejected by the Tellers.

Rule 6 The Tellers shall consider a ballot as valid provided the intent of the voter is clear, and provided also that he conforms with the regulations for voting.

Index to Constitution, By-Laws, and Rules of The American Society of Mechanical Engineers

Rubber Cushioning Devices

By C. F. HIRSHFELD,[1] DETROIT, MICH., and E. H. PIRON,[2] NEW YORK, N. Y.

This paper describes briefly the production of compounded rubber which is the material commonly used by engineers and others under the name of rubber. The wide range of physical properties obtainable by different compounding and different curing is indicated.

Springs of steel and springs of rubber are compared with respect to certain important characteristics and the conclusion drawn that neither can completely replace the other for springing purposes. There are applications in which one or the other gives the better or the only solution and there are other cases in which either may be used.

Various types and forms of rubber springs are described and illustrated and methods available for calculation and design are given.

Finally certain applications of rubber to the springing of rail vehicles are described and their results are illustrated.

RUBBER has long been known as a more or less mysterious substance of almost unbelievable potentialities. For generations scientists have experimented with it and industrialists have made a wide variety of products from it. Within comparatively recent years it has become an important engineering material.

The earlier uses tended to concentrate attention upon the chemistry and the simpler physics of rubber. The literature of the subject is voluminous but almost entirely devoted to such subjects. The engineer finds its pages exceedingly interesting but of little practical value to him in the application of rubber to his purposes except in a few highly developed fields.

It fell to the lot of the authors of this paper to be led into the use of rubber for mechanical purposes. They were forced to review available literature, to experiment with rubber in certain forms and for certain uses, and to cooperate with rubber manufacturers toward the production of designs and materials adapted to their needs. In this way they acquired a certain working knowledge of the material and of its technology, which information is made available in this paper. It is not intended as a profound discussion or a highly technical production. It is hoped that it may be useful in conveying to other engineers a sufficient over-all viewpoint to enable them to deal intelligently with rubber manufacturers in promoting the wider engineering use of this remarkable material.

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[2] Engineer, Transit Research Corporation. Mr. Piron was born in Belgium and was graduated from Louvain University in 1896. He has specialized in the theoretical study of new industrial problems and the application of their solution in practice in France, in Germany, and in Russia. He came to the United States in 1918. He joined the staff of the Transit Research Corporation in 1931.

Contributed by the Process Industries Division and presented at the Semi-Annual Meeting of The American Society of Mechanical Engineers, held in Detroit, Mich., May 17–21, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until October 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

What Rubber Is

The material known as rubber is derived from a milky liquid obtained by tapping certain plants, varying from trees to creepers and vines. This exudate is known as latex. It is a complex material consisting of minute globules of rubber, resins, proteins, various organic and inorganic substances, and enzymes, suspended in a liquid known as serum.

The rubber together with more or less of the other associated materials is separated from the serum by coagulation. This is produced in several different ways. When the latex is obtained from wild trees and other plants, the rubber is generally separated by dipping wooden paddles into the latex, holding them over a smoky fire until a solid-like material coagulates on the paddle, and then repeating this process time after time until sufficient amount of the coagulated material has been collected to form a merchantable lump or slab. When the latex is derived from plantation-grown trees, as is now generally the case, it is usually coagulated in flat pans by means of acetic acid and is marketed as sheets rather than lumps. These sheets are frequently washed and then dried in smoke houses at a temperature not in excess of 120 F. The smoking is continued for several days and results in preventing later fermentation of serum-borne materials trapped in the rubber during coagulation and not completely removed during the washing.

These lumps, sheets, and similar material in other physical shapes form the raw rubber of commerce. They are the starting point for the great bulk of the rubber industry. In recent years, however, many processes have been developed for making rubber products directly from the latex. In the state in which it is drawn from the plant, latex has a tendency toward rapid deterioration similar to putrefaction. This is inhibited by alkalies, and latex is now commonly prepared for shipment by mixing a certain amount of ammonia with it. Naturally it must be shipped in closed containers.

The use of rubber is exceedingly old in at least one part of the world. Archeologists have found rubber balls in Mayan ruins. It is believed they were used in a game and possibly also as sacrificial objects. In our own civilization the use of rubber may be said to have started late in the eighteenth century. It was then quite a curiosity. When it was found that it could be used for erasing marks by rubbing, it was given the name rubber by which we now know it.

In 1823 one Charles Macintosh established in Glasgow, Scotland, a factory for the production of waterproof coats and certain other products utilizing rubber. This is believed to have been the first industrial undertaking in the rubber field. It is interesting to note that this undertaking is still in existence in Manchester, England, to which city it was moved some time after its establishment.

The commercial use of rubber was decidedly limited until it was discovered that when mixed with sulphur and heated in the presence of lead salts it underwent a peculiar transformation. This invention is generally attributed to Charles Goodyear and assigned to the year 1839, although not patented until 1844.

The process was and is called vulcanization. It forms the foundation stone of the modern rubber industry. Very little rubber is used in what may be called its natural state. The great bulk is given a treatment corresponding to vulcanization. It is only through such treatments that the various properties required for different purposes and different applications can be imparted

to it. Rubber can be made to react with sulphur over a wide range of proportions. As more sulphur is taken up the properties change from those of the raw rubber through a complete series up to those of the hard rubber commonly known as ebonite. The most obvious effect of the reaction with sulphur is a stiffening of the rubber, using this term to mean a diminution of its tendency to behave somewhat like a viscous liquid and an increasing tendency to behave like an elastic solid. Hard rubber of the ebonite variety is obtained when something over 30 per cent of sulphur has reacted with or "combined" with the rubber. Commercial materials may contain any percentage of combined sulphur from this high figure down to as low as 1 or 2 per cent. Material of the type used for such purposes as are considered in this paper will generally contain nearly 5 to 10 per cent or even more.

A comparatively small part of the rubber goods of commerce is made by the simple process of vulcanization as originally used by Goodyear. Nearly all is compounded, that is, mixed with pigments and chemicals of various sorts in addition to the sulphur and the lead salts. Moreover, while sulphur is still the most commonly used vulcanizing agent, other materials, both organic and inorganic, have been found to produce similar results and are used for special products or special purposes.

Further, it has been discovered that vulcanization can be produced not only at elevated temperatures, as originally suggested by Goodyear, but also at room temperature by using certain gaseous and liquid compounds. It has also been discovered that many different materials act as accelerators, increasing the rate of reaction with sulphur and thus decreasing the time required for a given degree of reaction.

The material which is commonly spoken of as rubber and which is used for various purposes in our modern civilization is not the raw rubber as coagulated from the latex. It is the compounded and vulcanized material. Hereafter in this paper, the word "rubber" will be used to designate such compounded and vulcanized substances. The combination "raw rubber" will be used to designate the material as received by the rubber manufacturer for processing.

Technology of Rubber Processing

The technology of rubber processing is complicated in the extreme because of unavoidable variations in the raw material and because many different processes and many variations of individual processes are used to obtain specific results. However, there is what may be called a common technology used for the production of a large fraction of the manufactured rubber output. This is conveniently divided into steps as indicated below.

Washing and Drying. The raw rubber, particularly that obtained from wild plants, contains a certain amount of dirt and foreign matter as well as serum-borne materials trapped during the coagulation. It is customary to wash it with water and then to dry it to remove excess moisture. This washing is done in specialized equipment taking the form of wash kettles or wash mills. The drying is performed either in drying chambers at or near atmospheric temperatures or in vacuum-equipment. As has been stated previously, smoking is sometimes carried on in combination with drying.

Mastication and Compounding. The washed material is squeezed, pulled, and torn apart in some sort of mill. This converts it from a fairly tough, rough, and somewhat springy material to a plastic, tacky mass. In this condition it takes up readily the many different materials that must be mixed with it to produce the desired qualities in the final product. These consist of such things as various metallic oxides such as lead oxide and zinc oxide, carbon black in one of its many forms, sulphur, various vulcanization accelerators and, in many cases, antioxidants. The compounded rubber comes from these mills in batches and is ready for such forming processes as may be necessary before vulcanization.

Calendering or Sheeting. This or some equivalent process is commonly used after compounding. It is done by means of rolls which squeeze and roll the batch into long ribbons or sheets. These may then be cut into the desired shapes or they may be cut into pieces of convenient size for later processing. Variations of this process are offered by certain methods of extrusion by means of which long lengths of the desired cross section may be obtained.

Vulcanization. For this purpose the rubber is frequently placed in metallic molds which enclose it completely. The molds are then placed in presses or equivalent devices which hold them together with a certain predetermined pressure. While in this position the mold and its contents are carried through a temperature-time treatment which produces the desired degree of vulcanization. There are many variations of this step. It may even be performed without molds in some cases. In modern technology, the heat is nearly always supplied by steam and the temperature of vulcanization generally falls above 225 F and below 300 F. The amount of sulphur with which the rubber combines is determined by the percentage in the mix, the kind and amount of accelerator, the temperature, and the time under treatment.

Finishing, Inspecting, and Testing. When the vulcanization is complete the articles are removed from the molds or other devices in which they were vulcanized. Flashes resulting from flow into cracks or expansion chambers are removed if necessary. The articles are polished or otherwise finished if this is required. They are inspected for flaws of various sorts and, when necessary, for dimensions, and then tested to such extent as is required by the specification under which they are manufactured.

The technology just described is particularly applicable to such products as are indicated by the title of this paper. There are, however, many cases in which it must be varied to a very great extent. This is well exemplified by cases in which compounded rubber is used in combination with fabric or cords of some sort as in the manufacture of pneumatic tires and garden hose. In such cases additional steps are involved. There is another large group in which the cleaned raw rubber is put into solution and processed out of solution directly into the required shape. This is generally followed by some sort of vulcanization treatment.

Certain Properties of Compounded and Vulcanized Rubber

Comparison of the characteristics of rubber tubing, solid truck tires, and hard-rubber combs, all of which may be considered as compounded and vulcanized rubber, indicates immediately that wide variations of physical properties can be obtained. There are so many possible compounds and so many possible variations in their treatment that it is indeed difficult to make sweeping and yet truthful statements of physical properties. For the purposes of this paper it is sufficient to restrict such statements to such materials as are used for vehicle springing and other similar mechanical purposes.

Limiting the application to such materials, it may be said:

Rubber offers comparatively small resistance to change of shape under the action of small external forces. In the common or colloquial form of expression it is therefore said to be highly elastic. In the technical sense, the modulus of elasticity is of small numerical value. For example, when strained in tension a comparatively small force is required to double its length. Values of the modulus of elasticity, E, of the order of 100 to 400 are usual in comparison with the 30,000,000 characteristic of steel.

It is unique among materials used by the engineer in that it permits deformations of relatively tremendous magnitude. In the case of steel, extension within the elastic limit is exceedingly small. In the case of rubber, extension to double the original

length is a common occurrence and it is a fact that many compounds can be stretched to five or six times the original length, some to ten times. They do not even approximately obey Hooke's law in any part of the deformation; but, on the other hand, they are capable of complete recovery to the original length after the load is removed unless permanently damaged by incipient rupture. Its modulus of elasticity is low for tension, compression, and shear. It follows that it is deformed through relatively large distances by comparatively small forces in all three possible methods of loading.

In spite of its low compression modulus, it is an almost incompressible material in the same sense that water is. Most compounds of the types here under consideration are only slightly more compressible than water under ordinary conditions. In fact some stocks are less compressible than water up to pressures of 10,000 to 20,000 lb per sq in. This means that if completely enclosed it does not yield appreciably to compressive forces. As will be shown later, there must be free rubber surfaces to permit any great deformation under compression.

It has to a great extent the characteristics of a liquid, for which Poisson's ratio would be 0.5. The value of this ratio for rubber approximates closely 0.5 for small deformations. For very large deformations there is a marked departure from this value indicating that the material departs more and more from the true liquid condition. When a sample is tested in tension and when Poisson's ratio is determined in terms of the original dimensions, its value may vary from something of the order of 0.48 at the start to as low as 0.125 or 0.130 at an elongation of 500 to 600 per cent.

When held under a constant load of the magnitude used in practical engineering work, compounded and vulcanized rubber undergoes a continuing deformation. The rate of deformation is rapid at first and then decreases gradually with time. Tests extended over periods of years indicate continued deformation at an ever-decreasing rate. There is no evidence that a condition of equilibrium would ever be reached. This continuing deformation under load has been called both "cold flow" and "drift." This will be considered at greater length in a later section.

It follows that the shape of the stress-strain graph obtained when testing rubber will vary with the rate at which the load is applied and with the total time during which the rubber is held in a strained condition. A very rapid application of load and an early release will produce the minimum deformation because of the comparatively short time during which drift can occur. Larger deformation for a given load can be obtained by slow application and long duration.

The stress-strain graph obtained when unloading from such loads as are used in engineering work does not coincide with that obtained when loading. The graph obtained during unloading always subtends a smaller area than that obtained during loading. Interpreting the area under the graph obtained during loading as representing the work done on the material, it follows that it does not return all this during unloading. As a matter of fact, the temperature of the rubber increases as rubber is successively carried through load-and-unload cycles, indicating that at least some of the energy not returned is converted into heat within the mass of the rubber. It appears that the greater part of the energy that "disappears" is accounted for in this way. This phenomenon is known as "hysteresis" or "elastic hysteresis" because of its general resemblance to magnetic hysteresis.

There is another and closely associated phenomenon known as "elastic aftereffect." When a deformed specimen is unloaded it does not immediately return to its original dimensions. The loop obtained when a complete stress-strain cycle is plotted does not close. If the piece is allowed to rest in the unstrained condition it will ultimately return to, or very close to, its original dimensions unless it has been permanently injured. The amount of the elastic aftereffect and the length of time required for recovery vary tremendously with the type of compound and with the type of the loading and unloading cycle. Recovery may occur in a few minutes or in a few days depending on conditions.

A further phenomenon which may be designated "accommodation to load" is of importance. If a sample of rubber is loaded and then unloaded as during normal testing in tension, compression, or shear, and if this is repeated several times in succession each successive cycle will usually give results different from those of the preceding cycle. The rubber will be stiffest for the first cycle and successively less stiff for each succeeding cycle. The hysteresis loss will be greatest for the first cycle and successively less for each succeeding cycle. The elastic aftereffect will be greatest for the first cycle and successively less for each succeeding cycle. These differences will ordinarily persist to a measurable degree for from two to ten cycles, and sometimes for even a greater number. After that each cycle of loading and unloading will duplicate its predecessor provided the working of the material is not sufficiently severe to cause a continuing rise of temperature.

The elastic properties of rubber compounds are very sensitive to temperature. In the range between 0 F and 120 F, which for practical purposes is a comparatively narrow one, they frequently vary widely. The relations are quite complicated and exception can be found to almost any sweeping statement. In a general way it may be said that under increasing temperature in the range indicated: The rubber becomes softer, in the sense that a given load causes a greater deformation; the hysteresis loss decreases as though less internal friction had to be overcome; and the drift increases.

The difficulty of making simple and sweeping statements with respect to rubber is well-illustrated by the first of these three. The statement is true in a gross way. That is, a load-deflection or stress-strain graph for one compound at say 80 F will lie below that for the same compound at 40 F when stress is plotted as ordinate and strain as abscissa. But, there is a curious and superposed phenomenon. If a piece of rubber supports a given weight in tension and suffers a particular elongation as a result, it may become longer or shorter if its temperature is raised. With most mixtures at very low degrees of extension, the rubber appears to lengthen with increasing temperature but over the greater range of extension it will shorten under such conditions. Under the latter circumstances the rubber actually becomes stiffer with rising temperature in the sense that a given load causes a lesser deflection at the elevated temperature. Such phenomena make it very necessary to measure and record the temperature of test specimens when dealing with rubber and also the rate of loading and unloading.

By long usage it has become customary to test rubber in tension in the same way that it has become customary to test steel in tension. Accumulated experience makes it possible to judge many of the properties from the tensile test in both cases. The stress-strain graph obtained when testing steel varies with the character of the metal but always has a characteristic shape. When stress is plotted vertically the graph starts at zero with a straight line making a large angle with the horizontal. This gradually curves over to produce a curve running roughly parallel to the axis of abscissa and may even drop toward that axis before final rupture of the test specimen.

In the case of rubber there is also a characteristic shape of stress-strain graph in tension but the possibilities of variation of this material are so great that the characteristic shape is not always attained. The shape that may be called the normal for rubber compounds is given in Fig. 1. Near the origin there is a very small part of this graph showing a rough approximation to proportionality between stress and strain but it curves over

gradually to give a long smooth curve at a lesser angle to the axis of abscissa and then turns up again, showing a marked stiffening in the sense that large increases of load produce comparatively small deformations. The engineer accustomed to dealing with metals should note that the strains or deformations indicated frequently extend to beyond 600 per cent of the original length of the test specimen.

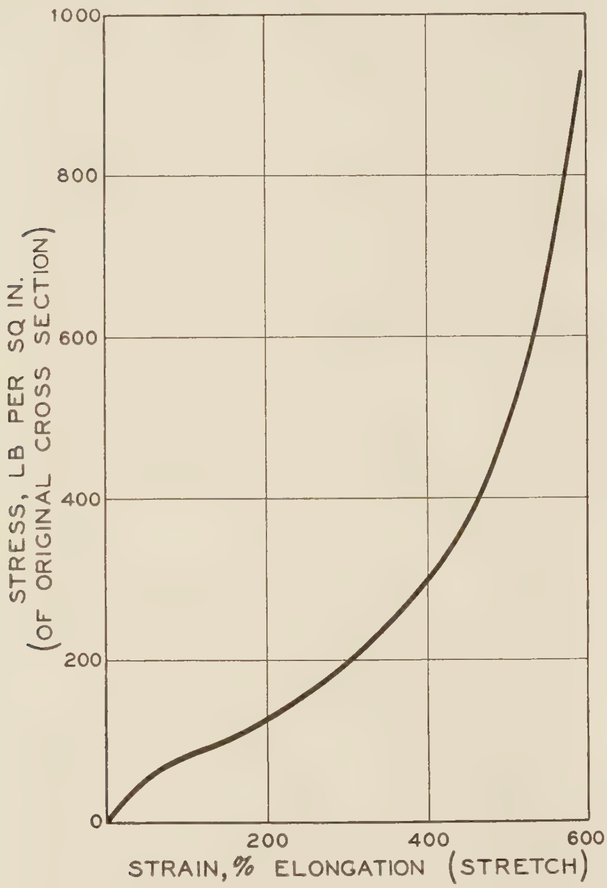


Fig. 1. Typical Stress-Strain Graph for Compounded Rubber in Tensile Test

This graph does not really tell a complete story because stress is plotted in terms of original cross-sectional area, as in the case of metals in ordinary practice. There is a marked necking effect during tensile testing. It is somewhat similar to that obtained when testing ductile metals. If the stress is figured for the actual cross-sectional areas assumed as the test progresses, the graph becomes much steeper than when put in terms of original area. A magnification of 5 to 8 times in the vertical direction (depending on degree of elongation and the value of Poisson's ratio) would not be uncommon.

In Fig. 2 are shown various forms that may be taken by the stress-strain diagram for rubber in tension. The variations may be produced by different methods of testing, different constituents in the mixture, different methods of vulcanization or cure, different temperatures, different water content, and different elastic history immediately preceding the test.

The peculiar significance of the modulus of elasticity in tension as used with metals, when used with respect to rubber, may be gleaned from the data indicated in Fig. 1. If the initial slope of the graph be used to calculate E, the modulus will be found to be about 120. If, on the other hand, corresponding stress and strain values along the length of the graph be taken, the results tabulated in Table 1 are obtained.

TABLE 1 MODULUS OF ELASTICITY FROM FIG. 1

Load (stress) lb per sq in., original	Elongation (strain) per cent, original	E
55	50	110.0
82	100	80.0
127	200	63.5
198	300	66.0
295	400	73.8
490	500	98.0
975	600	162.5

It is evident that the methods of calculation generally used in connection with metal tests produce widely varying values of the modulus for a given specimen of rubber tested in tension. Failure to take account of the varying cross section of the rubber is at least partly responsible for the variation. It would be better to speak of the "apparent modulus of elasticity" or the "instantaneous coefficient of elasticity" or some other indicative term when referring to values for rubber.

An example of the variation that may be met when different compounds come under consideration is furnished by comparing the results shown in Table 1 with those of Table 2 which apply to

TABLE 2 MODULUS OF ELASTICITY

Load (stress) lb per sq in., original	Elongation (strain) per cent, original	E
80	25	320
145	50	290
285	100	285
490	150	327
800	200	400
1070	225	475

a different grade of rubber tested in tension. On the basis of the slope at the lower end of the graph, E for this material is 500.

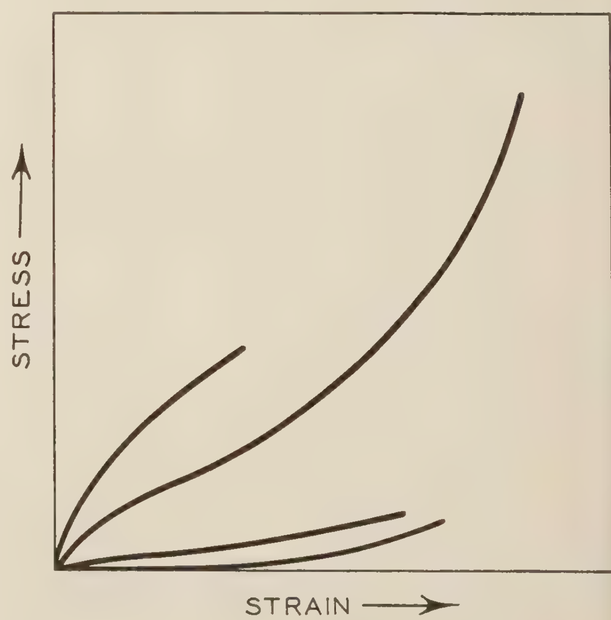


Fig. 2 Variation in Shape of Tensile Stress-Strain Graph for Compounded Rubber

It will be observed that in both cases the value of E first decreases and then increases. This must be true for all materials giving a stress-strain record such as that illustrated in Fig 1. It is only necessary, however, to look at some of the alternative forms shown in Fig 2 to realize that this is not a universal characteristic.

These apparent moduli cannot be used for purposes of exact calculation, as in the case of steel and certain other metals. They are useful for certain approximate calculations intended to determine roughly the sizes and types that may be used for certain purposes and within certain dimensions. The calculated results should always be checked on samples of similar size and shape where the attainment of given load-deflection ratios is important.

Compounded and vulcanized rubber of the type here under discussion is a poor heat conductor. The actual value of thermal conductivity varies considerably with the composition and treatment of the compound but is in the neighborhood of 0.1 Btu per ft per hr per deg F or lower, in comparison with about 26 for steel. It is of about the same order as that for cardboard and roughly about two to five times that characteristic of the common thermal insulating materials.

It is a poor conductor of high-frequency vibrations and is thus a poor conductor of sound throughout the audible range as well as of mechanical vibrations near the lower end of the audible sound range. This will be considered at greater length in a later section.

It has a comparatively high coefficient of thermal expansion. This also is quite variable depending upon the mixture and character of cure. But there is also another peculiar phenomenon. As the temperature is raised from very low subzero values a critical temperature is finally reached. At this temperature the coefficient of thermal expansion increases rather suddenly. The increase is frequently of the order of 100 per cent. In the case of mixtures used for springing purposes this critical temperature will nearly always be found below the minimum temperatures of use.

The cubical coefficient of thermal expansion for mixtures of these types will generally be found in the range of 0.00026 to 0.00033 per deg F. Roughly, this is about 15 times that of ordinary steel. This is of great significance in cases in which rubber is clamped between rigid metal parts which do not permit perfect freedom for change of volume with temperature, and the effect of temperature must be considered in all such designs.

It is highly resistant to many chemical substances although soluble in a number of the common organic solvents. Strictly speaking, rubber is probably the solvent but the end result is in all practical respects the same as though it were the solute. It absorbs readily a great number of organic oils and is very sensitive to the action of ordinary mineral lubricating oils. The latter are absorbed rapidly and "kill" the rubber in the sense that its elastic properties and even its cohesion are gradually destroyed. It is possible to compound and cure so as to obtain greater or less resistance to oil but the authors know of no compound which is at the same time oilproof and sufficiently resilient for springing.

It is readily attacked by oxygen and particularly by ozone unless protected by antioxidants worked into it during the compounding. Oxidation is most rapid at elevated temperatures and in the presence of ultraviolet radiation. Rubber exposed to direct sunlight is therefore apt to show rather rapid oxidation. Diffused light is not nearly so detrimental. When protected from light and not raised to excessively high temperatures, the rate of oxidation is comparatively low. Oxidation produces a hardening and a stiffening of the material and, if carried far enough, an actual cracking and gradual disruption.

Rubber readily picks up water in liquid or vapor form. It tends to maintain a rather complicated balance between the water contained within its mass and the vapor pressure in the surrounding liquid or space. Its physical characteristics differ with the water content However, such variations can generally be ignored in the ordinary engineering applications of rubber.

Compounded and vulcanized rubber tends to be isotropic in some cases and anisotropic in others. That is, in some cases it exhibits the same elastic behavior in all directions while in others this differs on the different axes of the piece. These properties are controlled by both the composition of the mixture and its treatment during fabrication. Certain compounding pigments, such as carbon black, zinc oxide, and lithopone tend to produce isotropic materials; others such as graphite, mica, and magnesium carbonate tend to produce anisotropic compounds. Calendering and extrusion both have a tendency to produce a "grain" and thus anisotropic characteristics. There is some tendency to eliminate during vulcanization the anisotropy produced by the latter method but such elimination is seldom, if ever, complete. In using rubber for springing purposes these characteristics are frequently of great importance.

Methods Used for Designating Compounds or Stocks

Each rubber manufacturer who makes a variety of products produces of necessity a great many different kinds of material. Variations are produced by varying the characters and quantities of the pigments, by varying the quantity of sulphur, by varying the character and quantity of accelerators, antioxidants, and other compounding ingredients, and by varying the temperature and time of vulcanization. The different materials are commonly spoken of as compounds or stocks and each manufacturer ordinarily uses key numbers and letters to designate the individual products. The word stock is however also used in a different sense to represent a class rather than an individual, thus tire stock, tubing stock, and hose stock are common expressions.

It is possible to produce a gradual modification of properties from one compound to another by an orderly variation of compounding materials or by an orderly variation of vulcanization or by a combination of these. Frequently this can be effected through wide limits in such a way as to produce a sort of family of compounds possessing similar over-all properties and yet with each physical property varying gradually as one ascends or descends in the series.

Many attempts to obtain a quick and ready means of testing rubber to determine its outstanding properties have been made. None has been completely successful. This should not surprise the engineer as he is already familiar with a similar situation with respect to other materials in common use. Thus in the case of steel he finds the yield point and ultimate tensile strength quite sufficient for some purposes. For others he finds it necessary to add extension and reduction of area. For others he requires impact tests in addition to the other values and for some purposes he needs hardness tests.

The same situation exists with respect to rubber. By long usage, the tensile test has come to be regarded as of great importance but, just as in the case of steel, it cannot be depended on to tell the whole story. In steel practice it has been found that so long as one deals with a known variety or with known varieties of steels, the surface hardness as determined with a Brinell or other hardness tester may be used to differentiate certain characteristics. Similarly in the rubber industry it has become customary to depend to some extent on an analogous test.

Two instruments are commonly used in this country for making such tests on rubber. One is known as the Shore durometer and the other as the Pusey and Jones plastometer. The former uses a spring-resisted needle to measure on an arbitrary scale the force required to produce a given degree of indentation of the needle. The reading obtained is known as the Shore durometer hardness or as durometer hardness or simply as the durometer of the rubber. Materials commonly used for springing purposes show durometer hardnesses between 30 and 70 or even 80 in extreme cases. The lower number indicates the softer material, that is, the material giving the greatest deflection for a given force.

The plastometer is a more elaborate instrument. It is designed to measure the indentation of a hardened steel ball of a given diameter when pressed into the surface of a rubber article or sample by a standardized force. The diameter of the steel ball is generally $^1/_8$ in. and the force is 1 kg. The indentation is measured in millimeters and is read directly from a dial. The softer rubbers used for vehicle springing and similar purposes will generally give a plastometer reading in the neighborhood of 175 and the harder in the neighborhood of 70 or less.

The indications of these instruments measure surface characteristics more than they do the physical characteristics of the interior of the mass of rubber. They are in some sense indicative of what may be expected of the mass when used on a series or family of compounds. They may not be at all indicative of the behavior of radically different stocks. Thus it is possible to find two rubber compounds with the same durometer hardness which have radically different physical characteristics in other respects and conversely, it is possible to find compounds behaving similarly in many respects and yet giving radically different durometer readings.

The unavoidable variations in commercial production result in a variation or spread of durometer readings for materials compounded and vulcanized as nearly alike as is now commercially possible. Further, the shape and size of the piece upon which the reading is taken also affect the indications of the instrument. As a result of these facts it is customary to allow a variation of about ± 3 durometer degrees in compounds of 30 durometer hardness and to increase this range as the material becomes harder until a tolerance of ± 5 is reached at a durometer hardness of 80.

The Aging of Rubber

Rubber compounds are not completely stable materials under ordinary conditions. Engineers are accustomed to the gradual wasting or deterioration of steel and other metals by corrosive attack. They are also accustomed to a change of physical properties with time, as in the case of certain aluminum alloys. The changing of rubber products with time is an analogous characteristic which the engineer must take into account if he chooses to use rubber in his work.

In a general sense, rubber ages by two entirely different methods or processes. One is the continuation of vulcanization and the other is oxidation.

Rubber products as delivered by the manufacturer are never completely vulcanized. That is, they always contain sulphur which has not reacted with the rubber. Reaction between the rubber and this surplus of sulphur appears to continue indefinitely. As one of the effects of reaction with sulphur is to stiffen the rubber it follows that this self-continuing vulcanization tends to produce an ever-stiffer material. However, with the types of compound used for the springing of vehicles and like purposes the aftervulcanization usually can be ignored for practical purposes.

Aging by oxidation is a far more serious matter. The first evidence of such aging usually is surface cracking or checking of the material. If the process continues far enough the mass begins to become checked and brittle so that it can be broken apart rather easily. The action ordinarily progresses inwardly from the surface in a fairly uniform way so that there is no danger of sudden failure. The progress of the disease is evident for months or even years before a dangerously large portion of the cross section has become affected.

Oxidation is greatly stimulated by the ultraviolet radiation in sunlight and by similar radiation derived from any other source. It is believed that the oxidation is produced by ozone which is formed from oxygen in the atmosphere under the action of ultraviolet radiation. Diffused daylight has very much less effect in stimulating oxidation than does direct sunlight. It appears that although oxidation of the surface is much more rapid when under ultraviolet radiation, the products formed offer some measure of protection to the interior of the mass and thus prevent the rapid spread of oxidation into the interior. On the other hand, rubber oxidized in the dark suffers a much more complete oxidation although it occurs at a much less rapid rate with all other things equal.

It is obvious that rubber should be shielded from direct sunlight and other sources of ultraviolet radiation to the extent that other considerations permit. It is also wise to protect it from the impingement of diffused daylight to the extent that convenience in other respects allows.

Experience has shown that painting the surface of rubber articles with certain protective paints, as for example aluminum in a suitable vehicle, tends to delay oxidation. It has also been found that certain synthetic rubber-like materials are not as sensitive to oxidation as are the natural rubbers. These are also effective in protecting the rubber when properly applied to its surface.

Finally, the rubber industry has developed materials known as antoxidants, antioxidants, antioxides and other similar names. These are mixed with the rubber when it is being compounded and serve to retard oxidation to a relatively tremendous degree. There are many theories regarding their action and much work remains to be done before this is completely explained. At least those materials in common use appear to be used up slowly as they protect the rubber. This indicates oxidation of the protective substance as a simple explanation of its effectiveness but it is probable that this is only a partial explanation at best.

The state of the art is now such that rubber with a relatively long life can be produced. The authors are unable to say what the expectable life may be under different conditions of use. They know of rubber springs that are still in successful operation under rather severe conditions after about eight years of use. They know of rubber that was used successfully in an enclosed position which was still usable for all practical purposes after fifteen years, although slightly checked on the surface. It seems perfectly safe to assume a life of five years for rubber springs that are not exposed to direct sunlight for long periods of time and are not exposed to attack by lubricating oil or other harmful material. The probability is that the useful life will be much greater than five years and possibly even greater than ten.

Drift of Rubber Under Load

This phenomenon was referred to briefly in an earlier section. It is believed to be of sufficient importance to justify more extended treatment. Some springs are used under such conditions that they are normally unloaded and that they assume and carry loads for a very small part of their total useful existence. Most springs are probably used under conditions requiring them to sustain a load throughout their existence, their function being to accommodate themselves to varying values of the load imposed. Vehicle springs all act in this way.

It is obvious that a spring which must sustain a load of magnitude during its entire useful life and which is made of a material which has a tendency to drift under load offers some interesting questions to the engineer. But, now that we are using steel at temperatures at which it is subject to plastic flow, the drift of rubber under load is not apt to cause as great concern as it might have a few years ago. The drift of rubber has been investigated to a sufficient degree to indicate that if due precautions are taken in the choice of compound and in the design of the spring it does not present any insurmountable difficulties.

As stated in an earlier section, the rate of drift under a constant

load decreases continuously although it does not appear ever to reach a zero value. The graph of Fig. 3 is drawn from results obtained with a tensile specimen. This specimen was made from rubber having a tensile strength of 4000 lb per sq in. of original section. It had a cross section of 1 in. by 0.138 in. and a length of 4 in. It was submitted to a constant load of 50 lb per sq in. of original section. Immediately after the application of this load the length was measured as 4.35 in. One hour later the length had become 4.40 in. The history out to the end of 168 hr is given in Fig. 3. It will be observed that the graph indicates an ever-decreasing rate of drift but gives no indication of its ever reaching a zero value.

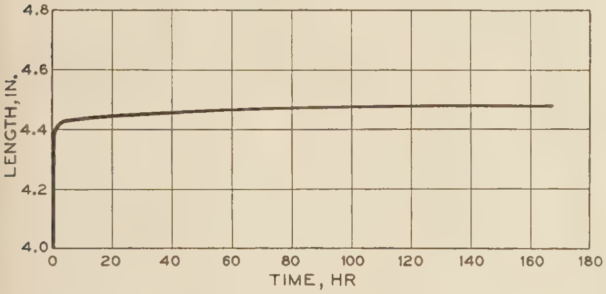


Fig. 3 Drift in Tension

It will be shown later that rubber is seldom used in tension for springing purposes of the type handled by the engineer. It is used much more often in shear or in compression. Most of the applications thus far made by the authors have used the shear deformation for springing for reasons which will be explained later.

Three graphs of shear drift are given in Fig. 4. The performances of three different materials are indicated. In their other properties they are not nearly as different as these drift performances might lead one to believe. It will be observed that at the end of 100 hr test, material *B* has drifted over three times as far as material *A*, and material *C* has drifted even more than material *B*. This indicates the importance of choosing compounds well adapted to the purpose in hand.

It will be recognized that these graphs have the same characteristics as that already given for tension but that the tension graph shows much greater drift. This could, of course, be due to the use of a different compound, to testing at a radically different temperature, to testing at different specific loadings, or to a difference in performance when tested in tension and in shear. The authors believe the results shown represent to some extent a real difference in the inherent behavior of rubber in tension and shear, respectively.

The drift results presented were obtained on compounds of the general type used for rubber shear springs. When it is realized that the spring tested for nearly 20,000 hr showed a total drift of less than 0.08 in. in a period of something over two years, and that during all that time it carried a load of the same general order as would be used in vehicle springing, it will be appreciated that the phenomenon of drift need not be regarded as a serious handicap if properly cared for in compound and design.

Drift tests are difficult to perform accurately. First, it is necessary to load the piece in such a way that static friction of bearings or other supports cannot influence the results. This indicates directly applied dead load or spring load as the most desirable. Second, it is necessary to arrange to read movements to 0.0001 in. or less if the course of drift is to be followed accurately. Third, it is necessary to control rather carefully the state of vibration of the piece under test. If the piece be absolutely free of vibration the progress of drift will be seriously interfered with and then very large movements will occur if accidental vibrations occur. Even the vibration of a building produced by moving machinery hundreds of feet away may be sufficient to cause erratic behavior if the machinery runs during some hours and not during others. It is as though a continuous movement of the mass of rubber is required to permit continuous and continuing readjustment of its molecules. The results graphed for material *A* in Fig. 4 were obtained with a piece kept under continuous vibration by means of a small motor rotating at 1750 rpm and carrying an eccentrically located weight. Fourth, the results indicated are exceedingly sensitive to temperature. In the case of material *A* it was found that a change of temperature of 1 deg F produced a positive or negative drift of about 0.001 in. An increase of temperature caused a temporary recovery and a decrease of temperature caused a temporary increase of deformation. This phenomenon was referred to in an earlier section of this paper.

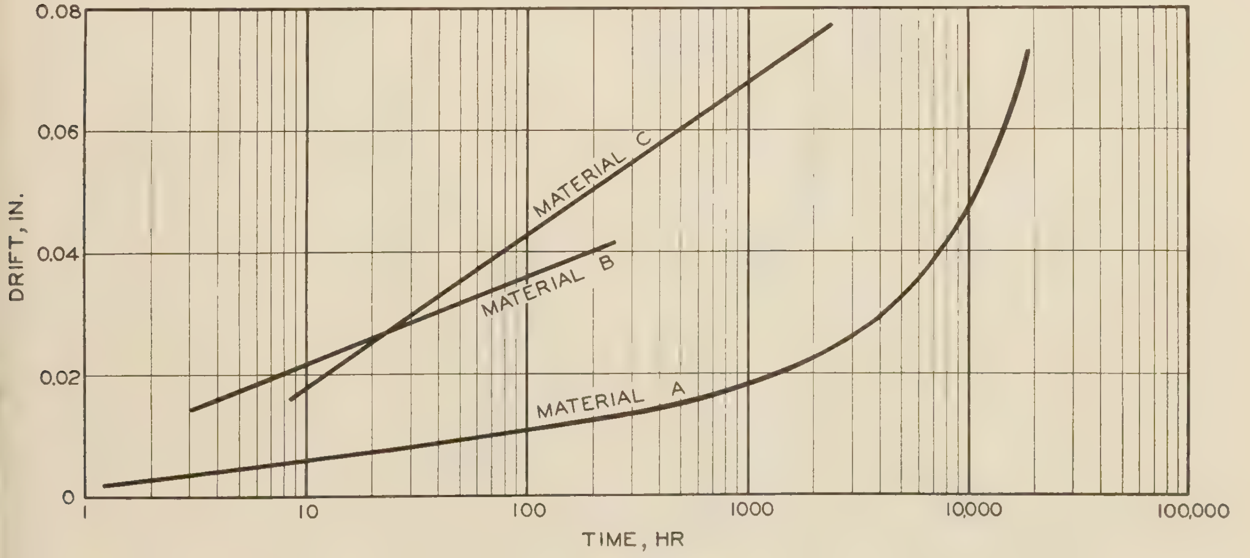


Fig. 4 Drift in Shear

Rubber as a Springing Material

As rubber has come into use as a springing material and has begun to displace steel and other metals for such purposes there has developed an expectable but regrettable controversy between the suppliers and the advocates of the two materials. This is regrettable because the arguments presented frequently have been partisan and therefore have tended to obscure the real facts and to delay impartial analysis. A case can be made for either material by properly choosing the assumed conditions and thus concentrating attention on this or that property or combination of properties.

It may be said categorically that rubber cannot completely displace metal as a springing material. But it is equally true that for certain conditions rubber is far superior to any known metal for springing purposes. The task of the engineer is the determination of the conditions under which rubber gives the better solution and the accumulation of the technical knowledge required to enable him to make the best use of rubber for conditions to which it is better adapted.

Examples of two methods of making comparisons between steel and rubber for springing purposes are given below. The first is based on actual values for the physical constants but on other assumptions which tend to give a result of little practical significance. It shows rubber as greatly superior to steel. The second compares two real and usable springs and the results would probably be interpreted as showing steel at least slightly superior to rubber.

For the first case we shall assume a steel wire, such as a piano wire, with a diameter of 0.02 in. and an elastic limit of 180,000 lb per sq in. The area of this wire is 0.000314 sq in. and at the elastic limit it can support

$$0.000314 \times 180{,}000 = 56.52 \text{ lb}$$

If its length be assumed as 100 in., it will have approximately an elongation of

$$\delta = \frac{180{,}000}{29{,}000{,}000} \times 100 = 0.62 \text{ in.}$$

when supporting the load of 56.52 lb. The potential energy accumulated in the spring will then be

$$V = \frac{56.52}{2} \times 0.62 = 17.52 \text{ in-lb}$$

The values just calculated indicate that the potential energy accumulated by this simple steel spring when loaded to its elastic limit is 558 in-lb per cu in. of material, or 1991 in-lb per lb of material.

For comparison with this we shall assume a piece of rubber which is 1 in. wide, $^1/_4$ in. thick, and 4 in. long. We shall further make the logical assumption that it can sustain without damage a tensile load of 2500 lb per sq in. and that under that load it will have an elongation of 500 per cent.

The energy absorbed by the rubber during stretching in this way could be determined as it has just been for the case of the steel wire if the stress-strain graph were a straight line. It has been shown that it is not a straight line and may be a rather complicated collection of curves. Strictly, one would have to integrate the area under the graph to obtain the value of the energy absorbed. For our present purposes it will suffice to assume the energy absorbed as two thirds of what it would have been had the stress-strain graph been a straight line. With this assumption we get

$$\text{Energy absorbed} = \frac{2 \times 1 \times 1}{3 \times 2 \times 4} \times 1 \times 2500 \times 4 \times 5$$
$$= 4170 \text{ in-lb}$$

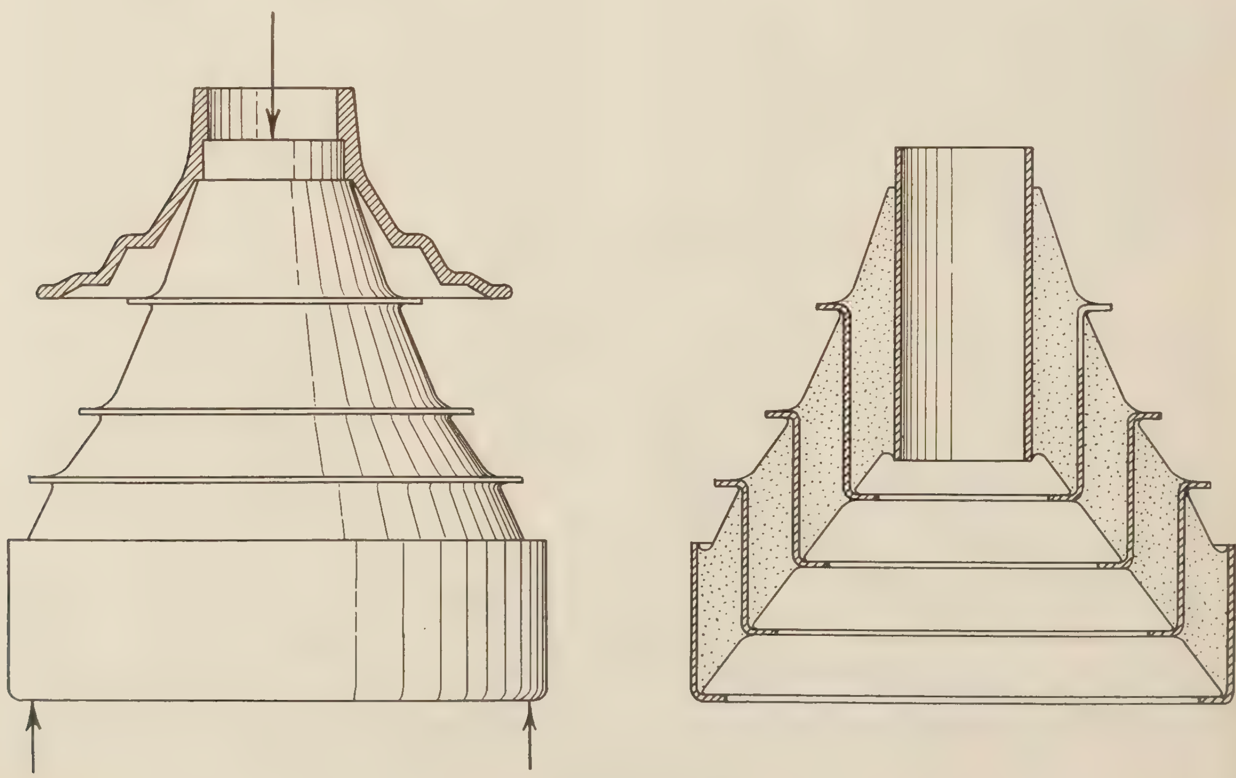


Fig. 5 Rubber Streetcar Spring

While this quantity of energy is absorbed it is not all converted into potential form in the sense in which we use that conception in connection with a metallic spring because of the peculiar behavior of rubber and its marked hysteresis loss. However, for present purposes, such refinements may be neglected and we may consider the potential energy stored in the spring as equal to the energy absorbed, in this case 4170 in-lb.

The volume of the rubber strip assumed is obviously equal to 1 cu in. in the unstretched condition. Its weight would be about 0.037 lb. With these values we obtain the potential energy stored in the rubber spring as 4170 in-lb per cu in. of material, or 113,000 in-lb per lb of material.

Evidently under the conditions assumed a rubber spring can store over seven times as much energy per unit of volume and over fifty times per unit of weight. Also the rubber spring of the type assumed would probably cost less than the steel wire. Apparently therefore, the rubber would constitute a far superior spring.

This is, however, not an entirely fair picture. It is true that rubber is occasionally used as a spring in a form analogous to that here assumed. The early use of rubber landing-gear springs on airplanes furnishes one example. The common "slingshot" of boyhood furnishes another. The rubber band used daily in offices and households is still another and probably the most common example.

But, steel is practically never used for springing purposes in the way that has been assumed. Its shape is changed to make it fit better into the assemblages in which it is to be used, although these shapes do not ordinarily make as good use of the physical characteristics of the material. It is used in such ways as to bring in its resistance to shear and to bending.

The fact remains, however, that for such purposes as have just been enumerated rubber does appear to possess certain unique advantages. It is difficult, for instance, to imagine an entirely satisfactory metal substitute for the common rubber band.

For the second example, we shall consider a rubber spring which is in actual use in the springing of streetcars and compare it with an equivalent coiled steel spring. The shape and structure of the rubber spring are shown in Fig. 5 and its load-deflection graph is given in Fig 6. The shape of the spring is essentially cylindroconical. The maximum-shear stress is limited to 60 lb per sq in. and certain refinements of design particularly applicable to springs of this type limit the distortion almost completely to shear strain. Bending strains, in particular, are reduced to negligible amounts by the insertion of the cylindrical steel partitions.

The volume of rubber in this spring is 180 cu in and it weighs approximately 6.7 lb. The combined weight of the rubber and steel is approximately 18 lb. The space occupied by the spring under a maximum load of 4000 lb is 410 cu in.

The load-deflection graph of this spring can be closely approximated by a nest of two helical steel springs. The outer one would have a free length of 13.2 in. or more, and a deflection of 4 in. under a load of 3300 lb. The inner would have a free length of 3.6 in. less than that of the outer helix and a deflection of 0.4 in. under a load of 700 lb. The resultant load-deflection graph is shown in Fig. 6, superimposed on that of the rubber spring.

The minimum possible weight of steel in this combination would be 28.43 lb although for practical purposes the design would be improved by using a somewhat greater weight of material. The volume occupied under a maximum load of 4000 lb is 250 cu in.

Inspection of Fig. 6 indicates that the quantities of energy absorbed by the rubber and steel springs up to 4000 lb load may be considered as equal. The area under the graph represents 4740 in-lb.

The comparison in energy absorbed between the two springs therefore becomes

	Steel spring	Rubber spring
Per cu in. of space	27 in-lb	16.5 in-lb
Per lb of material	237 in-lb	374.0 in-lb

If there had been included in the calculation the weights and volumes of spring housings required to contain the springs and to transmit the loads to them, the comparison would have been slightly more favorable to the steel spring on both the volume and the weight basis.

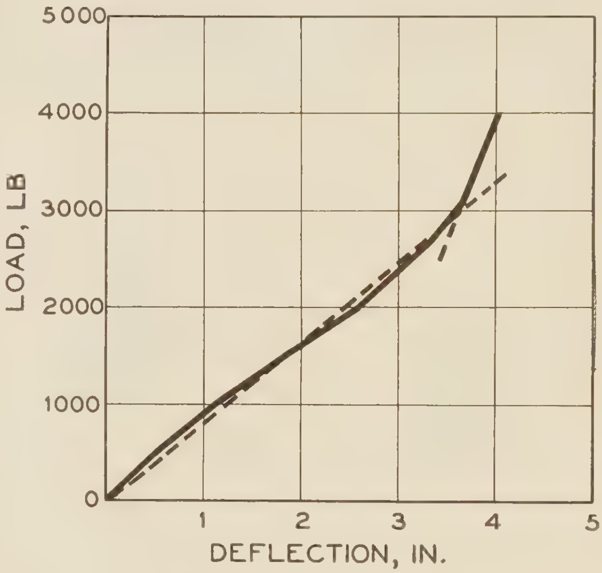


Fig. 6 Load-Deflection Diagram, Rubber Streetcar Spring

This comparison does not indicate any outstanding advantage for either variety of spring. However, a rubber spring of the type shown would probably cost more that the dynamically equivalent steel spring; certainly it is a more complicated structure; practically it is more difficult to test for fulfillment of specifications, and finally it does not have behind it the long period of successful use that the steel spring has. It is true that this is but one variety of rubber spring and that there are others of different types adapted to different purposes. Some are of less recent origin and much simpler in design, much easier to test and of comparatively low cost. However, it is a fact that no one has yet produced a design of rubber spring adapted to give deflections comparable to those obtainable with steel springs, which may not appear at first sight to be less desirable than the well-tried steel spring.

Under such conditions the engineer must find in rubber springs desirable characteristics not yet mentioned if he is to apply them in any case in place of metal springs. There are such characteristics and the authors, in common with many others, believe that some or all of these are sufficiently weighty in connection with certain applications to justify the use of rubber springs in such cases. A few of these characteristics are explained.

(1) It is common practice to use guides of some sort when parts are separated by metallic helical springs used in compression. This introduces frictional forces which are always so directed as to oppose the intended action of the spring. This is shown diagrammatically in Fig. 7 (*a*) and (*b*).

Such considerations are not of great consequence in the case of well-lubricated machinery. They may become serious however in other cases. An example is furnished by the common journal-box springing of many rail vehicles. Here exact fits are not ordinarily obtained or maintained and lubrication is bound to be more

or less imperfect. Moreover, as wear occurs and clearances develop, there results not only a less perfect guiding of parts but also the production of most objectionable noise.

With rubber springs it is easily possible to design so as to incorporate a nonfrictional type of guide within the spring itself irrespective of the design used. The guiding action is furnished by the rubber through its resistance to tensile, compressive, or shear deformation.

(2) Metals transmit audible vibrations, that is, sound, much more readily than does rubber. When used for springing purposes metals ordinarily are arranged so as to provide a complete metallic path between the place at which noise is generated and large metallic or other surfaces which serve as excellent broadcasting media. Attempts have been made to decrease the volume of sound transmitted in this way by using rubber cushions or other equivalent devices in the path. This has not been uniformly successful because the tendency is to use sound-insulating cushions which can be fitted into existing dimensions. The unit loads to be supported by the rubber cushions under such conditions frequently are so great that the cushion material cannot be given those physical characteristics necessary for maximum sound absorption.

It is, of course, possible to modify designs until the unit loads on such cushions have been reduced to the extent required to make them effective acoustic filters or insulators. But, when this is done the resulting volume of rubber will often be quite sufficient to provide all necessary springing if properly designed for that purpose.

The characteristics of rubber as a sound insulator have not been investigated to any great degree in a technical sense. In fact, the complications are so great because of the almost innumerable compounds and the consequent variation of practically all physical properties that comprehensive investigations would represent a tremendous undertaking. It is to be hoped that such investigations may be made in the near future for a few typical compounds of the types used in springing and similar mechanical uses. In passing, it is of interest to note that while the velocity of sound propagation in steel is of the order of 17,000 ft per sec, the velocity in rubber is of the order of 100 ft per sec. The relative acoustic-insulating values of the two materials cannot be determined directly from these figures because of the dearth of information regarding the mechanism of the elastic behavior of rubber. It seems probable that the ratio is even greater than indicated by the inverse ratio of these velocities. It also seems probable that with increasing frequency energy loss in the material increases at a greater rate with rubber than with steel. This is equivalent to saying that in comparison with steel, rubber becomes a relatively better acoustic insulator as the frequency of the sound wave or the pitch of the note increases. This is developed further in a later paragraph.

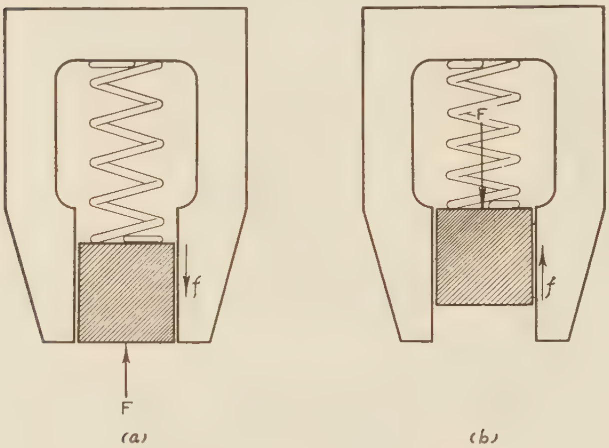


Fig. 7 Effect of Friction in Guides

(3) Multiple-leaf springs possess inherent frictional losses which perform a useful function as damping devices but nevertheless give such springs certain undesirable characteristics, particularly for vehicle springing. Two types of friction in these springs must be recognized, friction of rest and friction of motion. Both act to oppose the action of the spring somewhat as illustrated in Fig. 7 already referred to.

Leaf springs have been used extensively for vehicle springing because of their convenient shape, high load-carrying capacity, and inherent damping characteristics. But their use produces certain undesirable results which have not been as generally appreciated as have their virtues. The authors have performed some interesting experiments in this connection.

It is necessarily true that in the case of a sprung vehicle, impulses applied at the wheels must be transmitted to the body by means of the springs. If the body experience known vertical accelerations, it is possible to calculate the forces which must have been transmitted through the springs to produce such accelerations. If the load-deflection ratios of the springs are known, it is also possible to calculate the deflections which the springs should have suffered when transmitting the accelerating forces.

The authors measured vertical accelerations in a vehicle body and recorded at the same time the actual deflections of the leaf springs through which the accelerating forces were transmitted. Briefly, forces calling for spring deflections of 3 in. or more were transmitted with spring deflections of less than one third this amount. Obviously, in the short time during which the accelerating forces acted, the inertia and the friction of the springs caused them to behave more like solid bars of metal than like perfect springs. One way of expressing the result of this action is to say that there is transmitted to the vehicle floor a greater part of the impulse than would otherwise be received. Another way of expressing it is to say that the floor is subjected to greater amplitudes of vibration than it would be if the springing system were more nearly perfect.

It appears probable that the inertia of the spring parts and the friction of rest are responsible for a large part of this action. The impulses are of such short duration and reverse so frequently that most of the time of their action is probably consumed in overcoming inertia and the friction of rest between the leaves.

The explanation used may follow a still different course. If the friction of rest at the extreme position of vibration of the spring be neglected, on the assumption that there is always enough vibration and movement to prevent the attainment of a state of rest, there remains for consideration the approximately constant friction force between the relatively moving leaves. It is well known that damping of a system consisting of a mass and a spring by means of a constant damping force, such as this represents, results in a sudden change in the value of the reaction between mass and spring equal to twice the damping force at each reversal of direction of motion. These sudden changes of the reaction between mass and spring may then be held accountable for undesirable vibrations in the vehicle body.

In spite of the fact that rubber indicates a definite and sizable elastic hysteresis when tested and thus shows the presence of an inherent damping force, it is also a fact that this damping force reduces to zero value at zero velocity, that is at each reversal in the direction of motion. Thus the material is superior in this respect to the leaf spring. The hysteresis is of such a relatively small order in many cases that even with vehicles running on rails

it has been found desirable to provide damping devices to prevent continued oscillation under conditions favoring such action. Since such additional damping is not an inherent characteristic of the spring, it is possible to choose the most advantageous form of damping. This is commonly assumed to be viscous damping which is a function of the speed of motion. It assumes a zero value at each reversal of direction of motion and therefore is also devoid of the disadvantages of friction damping referred to above.

So much has appeared in the literature regarding the hysteresis loss of rubber that it seems desirable to enlarge upon this subject. It is certainly true that if rubber is, for example, stretched to several times its normal length, as is commonly done in testing machines, and is then allowed to return toward its original length, a hysteresis loop of large magnitude will be obtained when the stress-strain relations are plotted. A hysteresis loss of 30 to 40 per cent is not at all uncommon. A similar result will be obtained if the rubber is tested in compression or in shear.

However, if the loading and unloading are repeated time after time the properties of the rubber change with each repetition in such a way that a greater strain is produced for a given stress and the hysteresis loop becomes smaller and smaller. Undoubtedly it never reaches a zero value but with some compounds it may come so close to it as to make very accurate testing necessary to discover its existence.

The situation is complicated by several factors. One of these is time. If the deformation and recovery occur sufficiently slowly the mass of rubber cannot, in the limit, rise above ambient temperature and the process is an isothermal one. On the other hand, if deformation and recovery occur sufficiently rapidly, the process is, in the limit, adiabatic. The results are quite different in the two cases. In the former the hysteresis loss tends to remain of large value although such information as is available to the authors indicates that it does diminish somewhat with successive cycles. In the adiabatic case the tendency is toward a rapid diminution of hysteresis loss with succeeding cycles, apparently toward some limiting minimum-value characteristic of each compound. This limiting value is more theoretical than real because if an ideal adiabatic cycle could be performed time after time the rubber would be raised to such a temperature as to convert it by one means or another into a very different substance from that with which the experiment was started.

In practice it appears necessary to distinguish between two types of vibration, free and forced, and in the case of forced vibration, to distinguish between two different ranges of frequency when considering the damping characteristics of rubber springs, as used for example in vehicle springing.

The free vibration results when a single, nonrepetitive impulse causes a deformation of the spring and thus changes the configuration of the dynamic system. The latter then starts vibrating and the vibration dies out in proportions dictated by the type and extent of damping present. It is known that rubber-sprung systems show a rapid decay of vibration when subjected to single impulses in this way but nothing of an exact nature is known regarding the phenomena and laws involved.

In the case of forced vibrations one range includes the lower frequencies commonly encountered. In the case of vehicles these extend from about 0.5 to possibly 4 or 5 cycles per sec. The other range includes the higher frequencies, say from 20 cycles per sec upward. The omission of frequencies between 5 and 20 does not mean that they are not encountered. It is to be interpreted as meaning that frequencies between these limits appear to behave in an intermediate fashion, partaking of the nature of those above and below them. It is to be understood that no sharp boundaries can be set for any of the ranges in the present state of our knowledge.

Experience has shown that with rubber springs used in service which keeps them in practically constant motion, as in the case of vehicle springs, the hysteresis loss in the lower range of frequencies is comparatively small. That is, the energy loss in the springs is insufficient to provide the necessary degree of damping on rough roadbeds. They have therefore been used in combination with damping devices to prevent excessive amplitudes of oscillation under conditions yielding vibrations close to resonance frequency.

The story with respect to the higher frequency range is quite different. The continuous working of the rubber does not appear to reduce greatly the energy loss within it. It seems probable that there is some diminution of loss as the working proceeds and as an equilibrium temperature is approached. However, if there is such a diminution, there still remains a very large and effective damping force as an inherent characteristic of the rubber.

From fragmentary data, the authors are led to believe that the absorption of energy within the rubber in a nonreversible manner and within the higher frequency range increases approximately with the second power of the frequency. If an absorption of 10 per cent of the incident frequency happened to occur with a given thickness of a given compound at a frequency of 20 cycles per sec, one would then expect a loss of 40 per cent at 40 cycles per sec. Since the lower auditory vibrations are in the region of 25 cycles per sec, these figures indicate the very great effectiveness of rubber as an acoustic insulator.

Further, the authors are led to believe that the absorption of energy within the upper range and in nonreversible fashion increases with that dimension of the rubber which is effective in transmitting the vibration, at some power higher than the first. The increase appears, from fragmentary data, to follow a second-power law. If this be true, doubling the effective thickness should result in the absorption of four times as much of the incident energy.

With two second-power laws working thus in conjunction, it is evident that with high frequencies and thick rubber practically all the incident energy could be absorbed in the rubber. This appears to be substantially in accord with the facts for frequencies of 50 to 100 cycles per sec and rubber thicknesses of 4 to 6 in. These figures must be regarded as rough approximations because of all the variables which affect any given case.

Characteristics of Different Types of Deformation

Since rubber suffers relatively large deformations when acted on in tension, compression, or shear, it is possible to use it as a spring with any one of these methods of loading. However, each method has certain individual characteristics which fit it best for certain types of springing. It is therefore appropriate to consider these characteristics.

It has already been shown that rubber in tension has tremendous abilities as an accumulator of potential energy and that it is capable of astonishingly great extensions. This naturally fits it for use as a springing material in tension. However, it has several undesirable characteristics when used in this way. These have limited the rubber tension spring to special applications.

Most important is the great sensitivity of stretched rubber to physical injury. It is comparatively difficult to cut unstrained rubber. When strained in tension it is cut easily and a cut once made on the surface of strained material has a marked tendency toward autopropagation so that complete failure is apt to occur.

Again, rubber appears to drift to a much greater extent in tension than under comparable loads of other types and to have a greater elastic lag upon recovery. These characteristics can be explained as following directly from the shape of the rubber molecule and its behavior when loaded in different ways. However, the explanations are still quite controversial.

Further, Poisson's ratio has been shown to be of the order of 0.46 to 0.48 for low degrees of strain. If rubber may be assumed

to behave according to elastic theory the ratio between the modulus of elasticity in tension, and the shear modulus can be calculated from this ratio by means of the formula

$$G = \frac{E}{2\ (1 + \lambda)}$$

in which

E modulus of elasticity
λ Poisson's ratio
G shear modulus

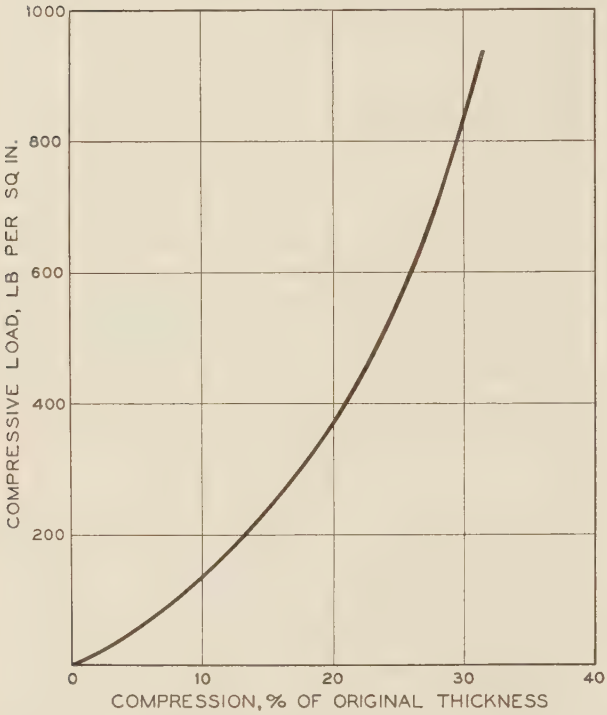


Fig. 8 Stress-Strain Graph for Rubber in Compression

It is evident that with $\lambda = 0.48$ the value of $G = \frac{E}{2.96}$ or approximately $\frac{E}{3}$. This is equivalent to saying that if the material obeys the elastic laws ordinarily postulated, a given deflection per unit thickness in shear can be obtained with about one third the force required to produce it per unit length in tension. Even with the lower value of λ encountered, about 0.125, the shear loading would still have a marked advantage.

Lastly, it is not easy to provide a means of holding rubber so that it can be used as a tension spring. The simplest method is to use a loop or a collection of loops as is done in the conventional rubber band and as was done in the early airplane landing gears. It would, of course, be possible to mold a dumbbell-shaped piece and to insert the enlarged ends in receptacles of the proper size. This may have been done but the authors have no record of any such attack upon the problem. They believe that with modern methods of "curing" rubber to metal, which will be described later, it would be possible to develop this and other designs for tension springs if it were desirable to do so.

In a general way it may be said that the use of rubber in tension for springing purposes should be confined to those cases in which the stretched rubber can be protected against surface injury and in which the spring is unloaded most of the time. The latter restriction appears to remove this form of spring from the field of vehicle springing and other cases in which a load must be sustained continuously by the spring and plus and minus variations of load accommodated. As a result of the limitations of the tensile type of rubber spring, it has become customary to do practically all rubber springing with the material in compression or in shear, or in some sort of combination of these two types of loading.

It has been shown in an earlier section that rubber subjected to tensile strain gives, in general, a typical stress-strain graph. It has also been shown that very wide variations from this typical graph may be produced by variations in the type of material tested.

Similarly, there is a typical form of stress-strain graph for rubber in compression and for rubber in shear. And, similarly,

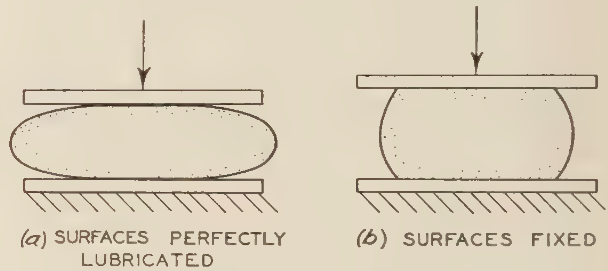


Fig. 9 Compression Loading; Same Total Load

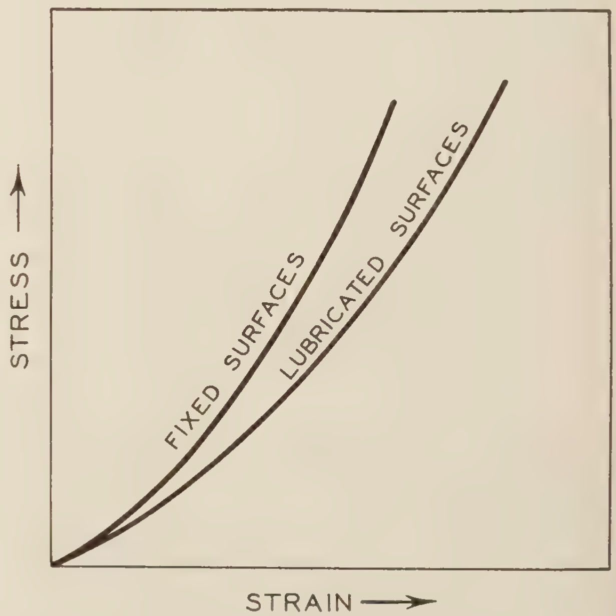


Fig. 10 Stress-Strain Graphs for Rubber in Compression; Fixed Surfaces and Lubricated Surfaces

very wide variations can be obtained by choosing appropriate compounds.

The typical compression graph is shown in Fig. 8. It will be observed that it is somewhat similar to that for tension loading in that the rubber stiffens as the load increases. As a matter of fact, for like material and in a general way, the stiffening in compression is at a much greater rate than the stiffening in tension. For example, the case shown in Fig. 8 indicates a deformation of only 32 per cent for a load of 900 lb per sq in. In tension, the

same material would probably show an elongation of over 200 per cent for similar loading.

There are two extreme possibilities in compression loading. These are illustrated in Fig. 9 (*a*) and (*b*). In Fig. 9 (*a*) the surfaces of the rubber in contact with the compressing faces are assumed to be perfectly lubricated so that they can slide along the compressing faces with negligible friction. In Fig. 9 (*b*) the surfaces are supposed to be relatively fixed so that no sliding can occur.

In the first case, the rubber acts as though it were less stiff than in the second case. That is, a given load produces a greater deformation. The stress-strain graphs would have the relationships indicated in Fig. 10, although there is no absolute significance in the relative values shown in the illustration.

Under laboratory conditions the perfectly lubricated requirement can be approximated, for example, by using a soap solution as a lubricant. Under conditions of practical use this is not possible and it is not feasible to use rubber with perfectly lubricated surfaces for springing purposes. It is of course possible to compress rubber between two metal surfaces without lubrication. This gives in effect a condition between those indicated in Fig. 9. The rubber slips to a certain extent upon the metal surfaces, particularly near the outer free surface of the rubber. The resultant stress-strain graph would lie between those shown in Fig. 10 and would approach one or the other more closely as dictated by the amount of relative slip.

Rubber is used for springing purposes to a great extent in just this way. A rubber slab of the required dimensions is merely fitted between two plane metal surfaces. With the normal grades of rubber and normal loadings the results are almost the same as though the rubber were actually fixed to the metal and did not slip with respect to it. That is, the graph would be almost coincident with that for fixed surfaces in Fig. 10.

There are, however, certain precautions to be observed when rubber is used in this way. Its behavior under increasing load is shown in exaggerated fashion in Fig. 11. It will be observed that there is a tendency for rounding of the edges in contact with the metal in the unloaded condition. If there are forces acting in the directions *S-S*, causing a shearing effect, there will be a sort of rolling action of the rubber near the rounded edges as the upper and lower surfaces undergo relative displacement. This tends to produce local deterioration of the rubber if it occurs to an excessive degree. The deterioration is probably due to a combination of abrasion and heating.

There is a growing tendency to produce a self-contained compression spring in which such is prevented. This is commonly done by what is known as "curing" or "bonding" the rubber to the metal faces between which it is compressed. For this purpose the metal faces are generally brass-plated, scrupulously cleaned, covered with a bonding cement and then placed in the rubber mold with the raw compound between them. The whole mass is then carried through the desired temperature-time curing schedule. When the operation is properly performed the bonding cement adheres very well to the metal on one side of it and to the rubber on the other side so that the rubber is effectively bonded to the metal.

It seldom happens that the metal parts between which a compression spring is to be located are so shaped and of such size that it is convenient to cure the rubber directly to them. For this reason there has been developed a structure known as a rubber sandwich. The rubber is cured between two comparatively thin pieces of metal and the design of the structure is arranged to receive the sandwich as a springing element. This is illustrated in Fig. 12 in which the sandwich is shown in place in a structure and under load as indicated by the bulge in the rubber.

When such bonds are well made they are capable of carrying relatively high shearing and tensile forces. This will be treated at greater length in a later section of this paper devoted to springs using rubber in shear.

The explanation of the radically different stress-strain performance of the same rubber tested in compression with perfect slip and no slip respectively between the rubber and load-imposing surfaces is comparatively simple. It has been stated earlier that

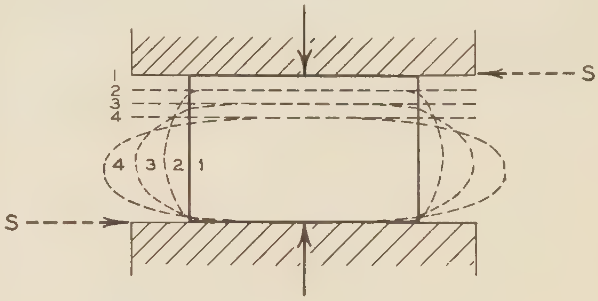


FIG. 11 RUBBER IN COMPRESSION, IMPERFECT LUBRICATION

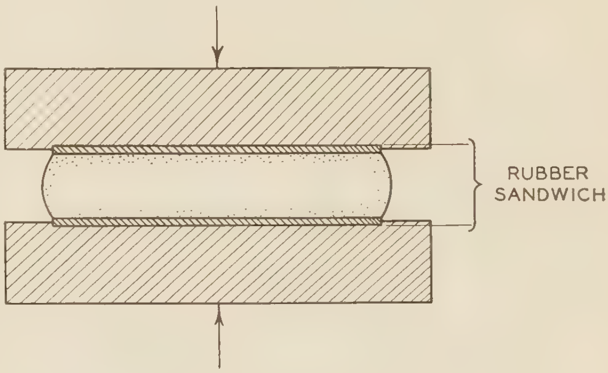


FIG. 12 RUBBER SANDWICH

rubber behaves like a liquid and, for practical purposes, is incompressible. Therefore, when loaded in compression it can yield in the direction of the compressive forces only as it can expand in other directions. Study of Fig. 9 (*a*) and (*b*) will indicate the significance of this. In the former the rubber is freer to expand laterally than in the latter. It is therefore capable of assuming a greater deformation in the direction of the compressive forces.

The modulus of elasticity for rubber in compression is just about as meaningless as is the modulus in tension. With perfectly lubricated surfaces one obtains different values depending on the extent of compression as indicated by the marked stiffening with load which is indicated in Fig. 10. Then as less and less slip is assumed the modulus for a given degree of compression becomes greater and greater, to reach an upper limit at zero slip.

In general, it can be said that for the conditions commonly met in rubber compression springs, the elastic modulus in compression varies from something around 200 to as high as 2000 or more. This value is determined not only by the character of the compound and its cure and the end conditions as referred to above, but also by the shape of the piece.

Some of the complications met in the design of rubber compression springs are illustrated in Fig. 13. Slabs of the same outside horizontal dimensions but of different thicknesses are seen to deflect very different percentages of their respective thicknesses under the same load. The thicker the material the "softer" it is as a spring. Cutting holes through it makes it still softer. It is

evident that there is some relation between the load-deflection ratio and the shape of the piece.

An extensive treatment of this subject was presented recently by W. C. Keys.[3] He relates certain areas which determine the behavior of the piece and thus obtains what may be called dynamic equivalents of different sizes. Using the expression load area to mean the area of one of the faces over which the load is applied, and the expression bulge area to mean the initial area of the free surface of the rubber, he calls the ratio of the former to the latter the area ratio.

In Fig. 14 (*a*) the load is assumed to be applied on the upper surface. The load area is then WL. The bulge area is the area of the vertical surfaces bounding the piece and is $2tW + 2tL$. The area ratio is then $\frac{WL}{2tW + 2tL}$ for this piece.

If the piece takes the shape of a short cylinder the load area is

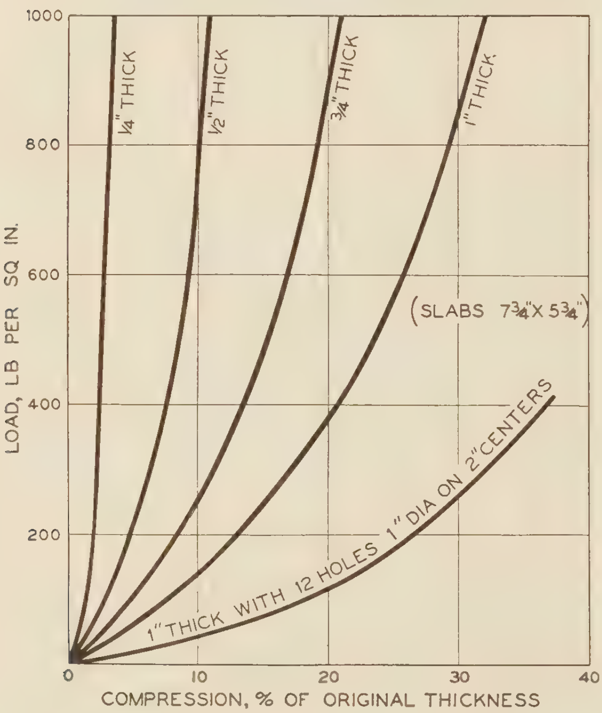


FIG. 13 STRESS-STRAIN GRAPHS FOR RUBBER SLABS IN COMPRESSION

obviously $\frac{\pi d^2}{4}$ and the bulge area $t\pi d$ in which t is the height of the cylinder.

It is interesting to note that when holes are made in the slab as shown in Fig. 14 (*b*) the surfaces of the walls of these holes are also included in the bulge area.

The paper[3] referred to contains instructions for the design of compression springs. The methods and data rest upon experimentally determined facts. They apply strictly only to cases in which the rubber is not bonded to the metal, is not lubricated, and is square in cross section. It has been stated that they can be applied to bonded or unbonded material and to rectangular sections and other sections.

The important result obtained, and which underlies the suggested design method, is "slabs of identical rubber compound having equal-area ratios and carrying equal-unit loads deflect the same percentage of their respective thicknesses." The paper contains graphs giving load-deflection data for different rubber compounds and for different area ratios with directions for using them for design purposes.

This paper by Mr. Keys is the first serious effort known to the authors to present orderly and authentic design information with respect to rubber springs. As such, it is a welcome advance over the fragmentary and often erroneous information hitherto available. However, actual use of the design methods outlined will show how much more experimental and analytical work is

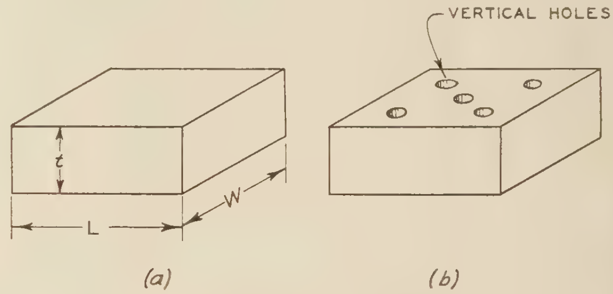


FIG. 14 COMPRESSION SPECIMENS

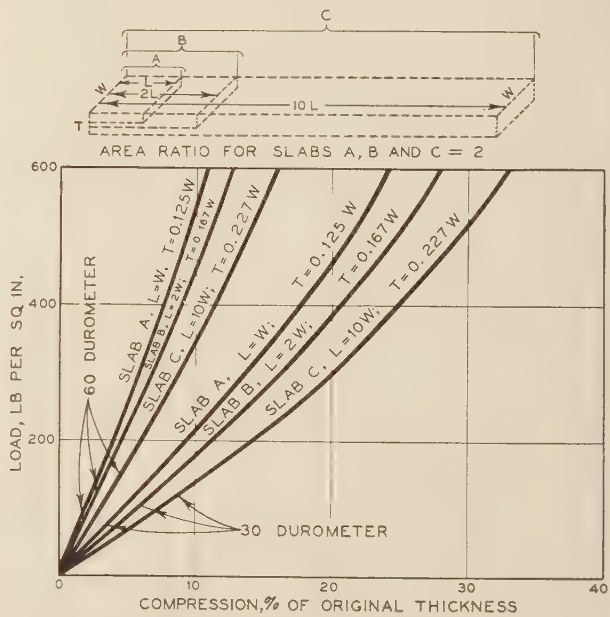


FIG. 15 VARIATION OF DEFLECTION IN COMPRESSION AS PRODUCED BY SHAPE OF SLAB

required to place the design of rubber springs on anything like the same plane as is that of steel springs. The author was limited to empirical methods of design because no one has yet rationalized the behavior of rubber.

An illustration of the extent to which results may depart from the general rule given by Mr. Keys is furnished by Fig. 15. Load-deflection values are given for three different compression sandwiches, all having the same area ratio. Values are given for two rubber compounds, one of 60 and the other of 30 durometer hardness. The variations in shape have been made extreme and the area ratio high to accentuate the departure. The departure from the suggested rule of design is extremely great, even at the low loadings.

This statement is not intended as a criticism of Mr. Keys's

[3] "Rubber Springs," by W. C. Keys. *Mechanical Engineering*, vol. 59, May, 1937, p. 345.

paper or suggested method of design. It is intended both to indicate that caution is still necessary in designing rubber parts on the basis of general simple rules and that there is a need for a great amount of investigational work in this field.

It is convenient to remember that for a given grade of rubber used in compression, the deflection for a given load increases as the area ratio decreases. In other words, the deflection increases as the relative bulge area increases. With the horizontal dimensions of the slab remaining constant, this can be accomplished by increasing the height. With the height remaining constant it can be accomplished by using a more elongated slab or a perforated slab when dealing with rectangular cross sections. With short, cylindrical slabs and constant height it can be accomplished by decreasing the diameter.

Certain characteristics of rubber determine the general practice used in connection with rubber compression springs. Drift or flow under load occurs just as it does in tensile loading. It is greatest at high unit loads and at high temperatures. For these reasons the unit static load is generally limited to a conservative value when dimensions or clearances are to be maintained. It is difficult to give generally applicable figures because of the wide variations produced by compounds and other variables. A permanent set or flow of 5 per cent of the original height is not uncommon with unit loads of 200 to 4000 lb per sq in. The upper limit of unit static load for conservative design is generally taken as of the order of 700 lb per sq in. Deflection is generally limited to from 10 to 20 per cent of the free thickness.

It is difficult to cure properly a slab of rubber which is too thick. This results from its poor thermal conductivity. The outside of a thick piece would be overcured (overvulcanized) while the inner part was still in a state of imperfect cure. For this reason, one dimension should be limited so that the central part of the rubber is not over a certain distance from an external surface. The maximum is generally regarded as about 1 in., giving a maximum total thickness of a slab as 2 in. Most rubber manufacturers appear to prefer to limit the thickness to 1 in. when possible.

The factors enumerated place certain limitations upon what can be done with compression springs. If the limit of thickness be taken at 2 in. and the maximum permissible deflection under static load as 20 per cent, it is obvious that a deflection of 0.4 in. is all that can be attained with a solid slab. With the preferred 1 in. thickness the deflection would be only 0.2 in. Greater deflection can be obtained by perforating the slab in the direction of the loading forces but this procedure is limited in its applicability. As the bulge area is increased by this method the load-bearing area is decreased so that the method is self-limiting.

To obtain greater deflection than can be attained practically with a single thickness, it is customary to stack two or more slabs. In some cases this is done with rubber in contact with rubber but it is now more common to stack rubber sandwiches metal to metal as shown in Fig. 16. To prevent sidewise slip between the metals various doweling arrangements are used. It is possible to produce a simpler arrangement by producing the stack as a single unit. That is, the intermediate metals have rubber bonded to both sides so that the doubling up of metals and the necessity of doweling are eliminated. The molding and other costs are necessarily higher so that this method is, in general, used only in the case of large production or to meet very special needs.

It should be noted that part of the total height in all such constructions consists of metal. This metal is generally at least 1/16 in. thick. Therefore with limited dimensions for accommodation of the spring, the presence of metal reduces the possible quantity of rubber and therefore the possible deflection.

It has been mentioned that rubber in compression is seldom used with static loads producing more than 20 per cent deflection. This is in terms of what may be called the thickness of the rubber. It is a fact that rubber may be used in shear with a deflection equal to the thickness. This has led some to conclude that a smaller amount of rubber can be used for a given springing job by utilizing its shear rather than its compression characteristics. The following examples will show that no such sweeping statement can be made.

Stress-strain calculations have been made for two square rubber sandwiches used in compression. One is 4 × 4 × 1 and the other is 6 × 6 × 1, all dimensions being in inches and the thickness being 1 in. A unit load of 225 lb per sq in. or a total load of 3600 lb produces a deflection of 20 per cent or 0.2 in. in the case of the 4 × 4 in. sandwich. For the 6 × 6 a unit load of 405 lb per sq in. or a total load of 14,580 lb is required to produce the same deflection.

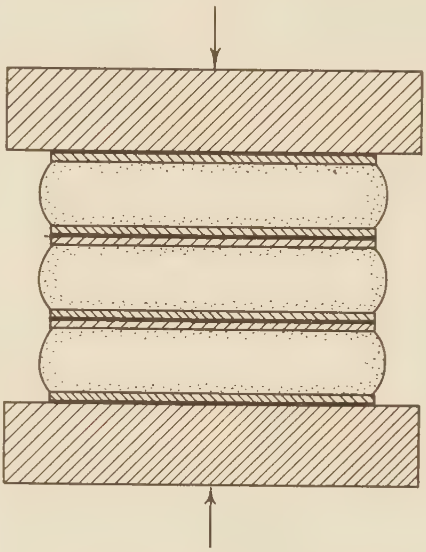


Fig. 16 Stacked Compression Sandwiches

In the case of sandwiches used in shear it has been quite common to limit the unit static shear stress to 20 lb per sq in. and most designers now prefer not to exceed 40 lb per sq in. Shear sandwiches have been proportioned on both these values to carry the same loads and at the same deflections as the compression sandwiches just referred to. The total volume of rubber required in the several cases is given in the following tabulation:

Total load, 3600 lb; deflection 0.2 in.
Volume, 4 × 4 × 1 in. compression sandwich, 16 cu in.
Volume, shear sandwich at 20 lb, 144 cu in.
Volume, shear sandwich at 40 lb, 32.4 cu in.

Total load 14,580 lb; deflection 0.2 in.
Volume, 6 × 6 × 1 in. compression sandwich, 36 cu in.
Volume, shear sandwich at 20 lb, 583 cu in.
Volume, shear sandwich at 40 lb, 131 cu in.

It will be observed that on this basis of comparison there is no case in which the shear sandwich uses less rubber than the compression type. In each case in which the shear sandwich is loaded at 20 lb the deflection is only 25 per cent of the thickness and in the case in which the 40-lb load is assumed the deflection is only about 56 per cent of the thickness. For the variations here under consideration the deflection in shear under a given load may be assumed to vary directly with the thickness. Therefore the shear sandwiches could be made to produce greater deflection by thick-

ening them. But the same sort of result could be obtained by changing the shape of the compression sandwiches, by thickening them, by stacking them, or possibly even by perforating them, so that the relations shown in the tabulation would not be materially altered by striving for a greater deflection.

The choice between compression and shear loading cannot be placed on any universally advantageous characteristic of one or the other type. The decision must be made as the best compromise in each case. Such things as available dimensions, directions in which the forces are to operate, freedom desired in directions other than that of the principal force considered, and others may all come in for study.

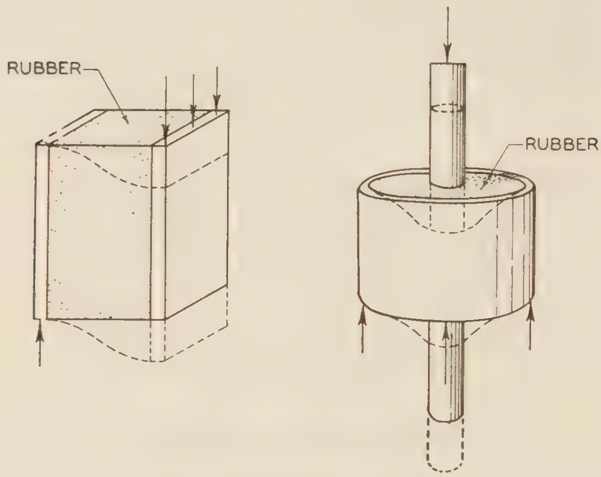


FIG. 17 STRAIGHT SHEAR SPRINGS

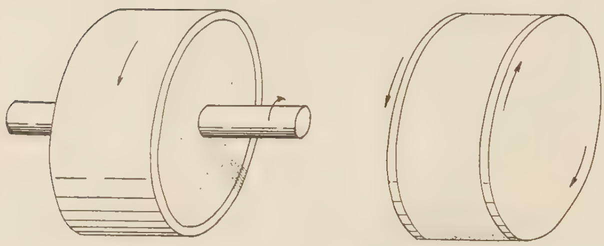


FIG. 18 TORSION SHEAR SPRINGS

For example, a compression sandwich is capable of functioning as a shear sandwich in all directions at right angles to the line of compressive force. It will function as a shear sandwich if there are forces at an angle to the line of the compression force and if the structure into which it is built permits. Again, a simple shear sandwich is capable of acting as such in two directions at right angles to each other and as a compression sandwich in a direction at right angles to both of these.

As another example, it may be convenient to suspend a heavy machine or engine by means of shear sandwiches fastened along the sides of the frame or underframe and most inconvenient to place compression sandwiches under the structure.

The use of rubber in shear as a springing medium is probably of most interest to the engineer because of the great variety of possibilities. These may be divided broadly into two classes which may be described as those in which the rubber is used in straight shear and those in which the rubber is used in torsional shear. Two simple examples of the first type are shown in Fig. 17 and two of the second type in Fig. 18. The use of rubber in shear was decidedly limited until satisfactory methods were developed for bonding rubber to metal. This made it possible to apply forces to metal and to transmit them to the rubber in a simple manner. Probably as a result of the late development of a practical means for using rubber in shear, the study of its behavior under such loading is far behind that of compression and tension loading. Work in which the authors have been engaged during the past several years made it necessary for them to study the behavior in shear to a sufficient extent to enable them to produce shear springs for certain limited purposes. They were hardly able to scratch the surface of this subject in the time and with the facilities available. But they did develop certain facts and viewpoints which may be of assistance to other workers in this field.

The simplest conception of a rubber sandwich acting in shear is shown in Fig. 19. The opposing forces produce a deformation as indicated by the dotted lines and the deformation is given by the equation

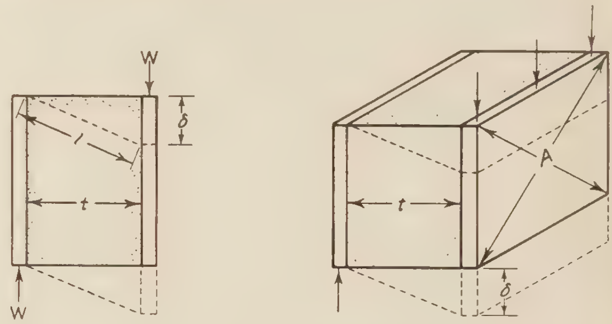


FIG. 19 RUBBER SANDWICH IN SHEAR

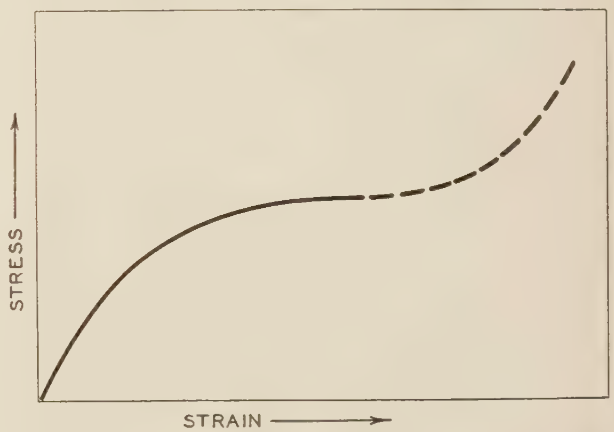


FIG. 20 CHARACTERISTIC STRESS-STRAIN GRAPH, RUBBER IN SHEAR

$$\delta = \frac{Wt}{AG}$$

in which

- δ deformation in inches
- W force or weight in pounds
- t thickness of the rubber in inches
- A shear area in square inches, and
- G shear modulus or elastic modulus in shear.

As in other cases already discussed, the modulus is not only a variable with respect to compound and cure but also with respect to shape and degree of deformation. The characteristic stress-strain graph for rubber in shear is shown by the full line in Fig. 20. It will be observed that it is substantially opposite in general

shape to the characteristic compression graph. It will also be observed that it indicates a varying apparent shear modulus. However, within the range of deformation in which rubber is commonly used in shear, the modulus for a given material in a given shape does not ordinarily vary more than 20 to 30 per cent. For different grades of material it varies from something of the order of 45 as the lower limit for the soft stocks to something of the order of 225 to 250 for the harder stocks.

The authors are of the opinion that if a shear test could be continued to sufficiently high deflections, the stress-strain graph would ultimately turn up as indicated by the dotted part of Fig. 20. There appears to be good theoretical reason for such behavior.

Inspection of Fig. 19 will show that the volume of the rubber does not change as shear deformation increases. But if a fiber of length t in the upper surface be thought of, it will be realized that it must be elongated to a length indicated by l. This sort of action appears to result in putting a tensile load on the bond between the rubber and the metal. In experiments in which rectangular sandwiches were loaded in shear in this way it was found that with sufficiently severe loading the bond let go at certain corners of the metal and that the fault then progressed gradually along the edges and toward the central area.

Such concentrations of stress at the corners could be diminished to some extent by rounding the corners and could be eliminated completely by making the sandwich circular instead of rectangular in shape as shown in Fig. 21. In fact, a sandwich of this general type is used in wheels developed by the authors which will be described later.

It will be observed that this sandwich has the same characteristics in any radial direction, and that these will be of the type illustrated in Fig. 20. However, it is comparatively stiff in a direction at right angles to its plane faces and tends to stiffen under increasing load.

The type of spring shown at the right in Fig. 17 has no corners but yet has different characteristics from those of the type shown in Fig. 21. Loaded vertically it acts in shear with a typical shear-spring characteristic. In all directions at right angles to the vertical it acts as a compression spring and with compression-spring chacteristics. With respect to three principal axes, the spring of Fig. 21 may be made soft in the direction of two and hard in the direction of the third. The exact opposite is true for the spring of Fig. 17.

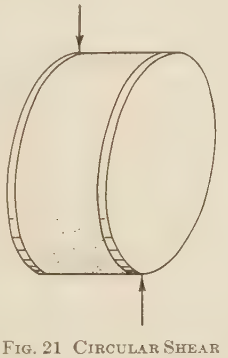


Fig. 21 Circular Shear Sandwich

The simple analysis given in connection with Fig. 19 is only a rough approximation to the truth. The shear spring shown in Fig. 22 will assist in demonstrating this. The thickness has been made much greater than in the earlier presentation. It is obvious that Fig. 22 represents a cantilever beam if, for example, the left end be assumed fixed. The deflection due to shear would increase uniformly from the left toward the right. But, as a cantilever, the deflection would increase with the cube of the distance from the left-hand toward the right-hand end. It is evident that no simple shear formula as suggested in connection with Fig. 19 can give correct results in the case of cantilever action. Moreover, pure cantilever action would require that the right-hand end of the beam in Fig. 22 rotate so as to remain perpendicular to the neutral axis.

In practical work with shear sandwiches the conditions are usually such as to prevent such rotation so that further complications are introduced. This is indicated in Fig. 23. The restraint offered by the guiding of the right-hand end results in the production of deflections as shown by the dotted lines, with high concentrations of tensile stress at a and b, respectively. It has been observed in tests that the separation from metal previously spoken of tends to occur at the upper corners and edges of the left-hand plate and at the lower corners and edges of the right-hand plate, with configuration as in Fig. 19. It appears probable

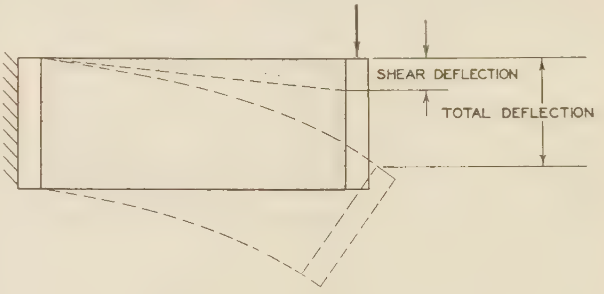


Fig. 22 Very Thick Shear Spring

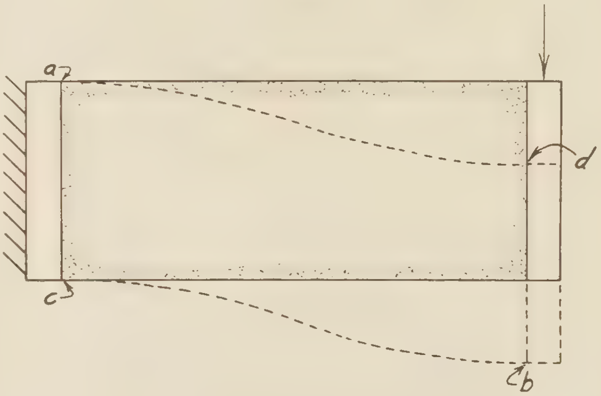


Fig. 23 Thick Shear Spring, End-Guided

that the beam action as just explained is responsible for this phenomenon.

Since the beam deflection varies with the third power of the thickness, its effects can be minimized by using thin sandwiches instead of thick ones. Experience shows no particular difficulties with rubber an inch thick unless exorbitant deflections are attempted.

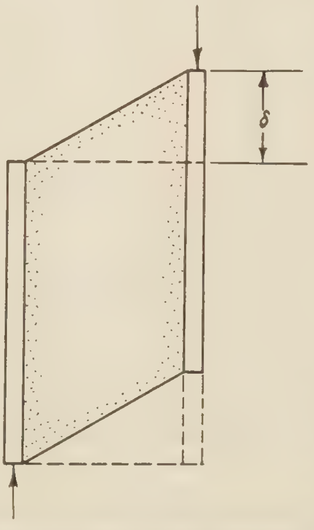
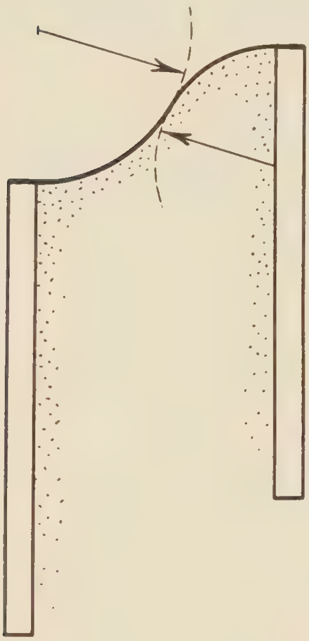


Fig. 24 Biased Shear Sandwich

It is sometimes possible to use a sandwich of the shape shown in Fig. 24. When such a spring is deflected, the length of the upper and lower surfaces tends to become shorter instead of longer up to the deflection δ when they reach the horizontal. By using proportions which will put all the deflection above the horizontal lengthening of the surfaces can be prevented and the tendency to pull loose at corners appears to be minimized. By choosing proportions such

that half the deflection produces the horizontal configuration, the lengthening of the surfaces will be approximately half what it would be with the initial shape as in Fig. 19.

The use of what may be called a biased sandwich is not an unmixed blessing. As distortion toward the horizontal direction occurs there may be a folding of the rubber in the upper or lower surface or both. If this occurs to too great an extent, the rubber will deteriorate rapidly at the place in which the fold appears. The laws governing this action are not known to the authors. It has been found by them and by others that folding can be prevented or minimized by changing the top and bottom from plane to curved surfaces of the proper shapes. At the present time the proper shape must be determined experimentally. It generally approaches the equivalent of two arcs joined by a tangent as shown in Fig. 25 although other and simpler shapes have been discovered recently.



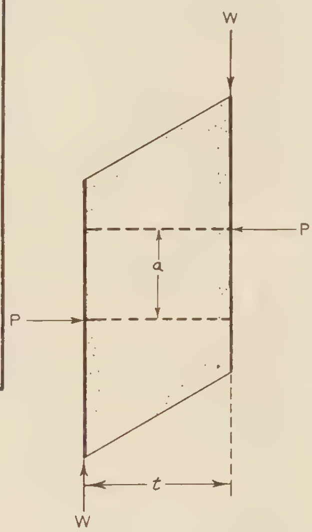


Fig. 25 Curved Top Contour Fig. 26 Balancing Moments

It has been found that crosswise compression of the sandwich tends to prevent separation of rubber from metal at critical points. This also is not a universal cure since the behavior of rubber like a liquid makes it possible for compression loads to produce tensile strains at unexpected places.

Precompression can be used for another purpose. This is illustrated in Fig. 26. It is evident that the load W directed vertically downward produces at the left-hand plate a moment Wt. For a given load and a given configuration, it is possible to choose a precompressing force P and a vertical distance between centers of application such that the moment Pa just balances the Wt. Since the deflection will change as W changes and, within limits, in direct proportion to W, it is not possible to maintain a perfect balance of moments over a wide range of load and with a constant value of P.

It has been proved experimentally under a wide range of conditions that the use of precompression does not alter the load-deflection ratio of a shear sandwich, at least to any significant degree. Precompression pressures vary from about 50 to 150 lb per sq in.

When testing sandwiches of different thicknesses in shear, there is some indication of a unique condition adjacent the plates to which the rubber is bonded. Fragmentary data appear to show that for the purpose of calculating load-deflection ratios with a given rubber, the thickness of active rubber should be taken as the actual thickness less twice the thickness of this unique layer.

It has already been indicated that the bonding of rubber to metal in a commercial sense is a comparatively recent invention. Most rubber manufacturers will now guarantee a bond strength of from 200 to 250 lb per sq in. It is not uncommon to attain a bond strength of over 400 lb per sq in. Some bonding methods now being experimented with are claimed to give a strength of about 2000 lb per sq in. In view of the many unknown factors still encountered in the design of rubber springs and in view of the newness of the art, it is customary to limit loads upon these bonds to values of the order of 25 to 30 lb per sq in. except in cases which have been very carefully explored and tested. Under favorable conditions the authors have used values as high as 50 to 60 lb.

Enough has been written to indicate that the design of a rubber shear or torsion spring upon the basis of simple shear action over the entire cross section of the rubber may lead to unexpected results. On the other hand, if experience is available with a given shape of spring in proportions not too different from those required, a value of the shear modulus derived from the existent springs may be used with simple shear and torsion formulas to design new springs. It must be realized that what is called the shear modulus in such cases is really a grand composite of a number of different factors which have been lumped into one

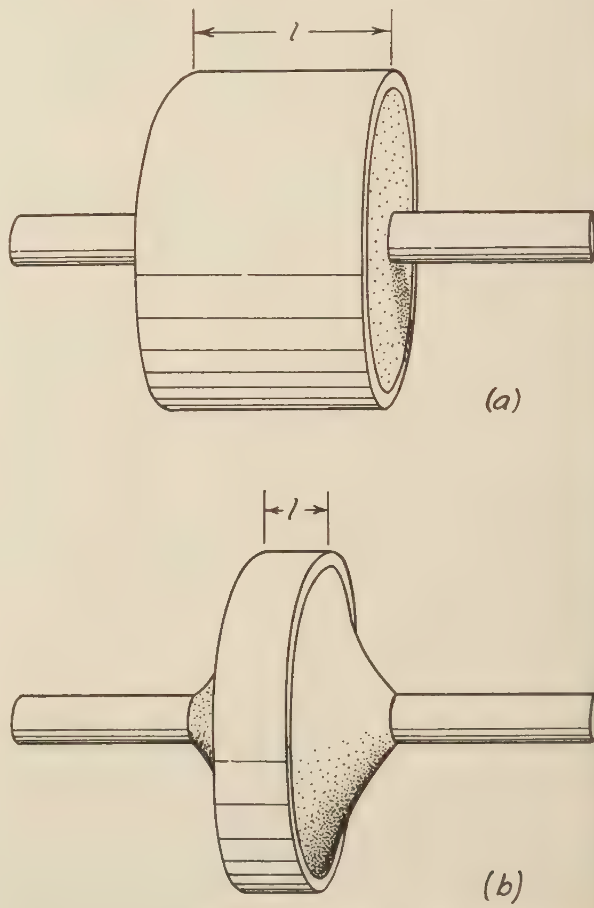


Fig. 27 Shear Springs; (a) Cylindrical Form, Nonuniform Stress, (b) Special Form for Uniform Stress

"constant." This constant can be used with assurance only for conditions which do not depart too radically from those for which it was determined.

Rubber is a comparatively expensive material. For this reason refinement of design which will decrease the quantity of rubber needed for a given purpose is frequently justified. An example is shown in Fig. 27. The spring indicated by (*a*) obviously makes poor use of the rubber distant from the central shaft. The maximum stress in the rubber occurs at the inner shaft and falls off with the square of the radius. The form shown in (*b*) takes this into account and varies the length *l* of the rubber to maintain a constant stress. Where many springs of one kind and size are to be built the extra mold costs that may be necessary for the more complicated shape required to save rubber will generally be found a paying investment.

Rubber Springs in a Streetcar

With the assistance of others, including research men, streetcar operators, streetcar manufacturers, and rubber manufacturers, the authors have developed during the past few years a streetcar in which rubber is used for several different springing purposes. These cars are now in commercial operation in a number of cities in this country. Two of the applications of rubber are described in the following paragraphs.

It should be emphasized that rubber was not used merely as a novelty. It was adopted after long and careful study and test because it was found to give a better solution that any other that was thought of. The practical performance has thus far been satisfactory and there is now no indication that rubber does not offer a successful means of accomplishing the ends desired.

One objective was the reduction of the unsprung weight to achieve the ends thus obtainable. Another was the reduction of noise made by the streetcar. It was felt that the first objective could be obtained by the use of a resilient wheel, and the second could be furthered by using rubber as the resilient material. After the trial of many semicommercial designs and the development and trial of many designs of our own, the one shown semidiagrammatically in Fig. 28 was adopted. It is now in use on several hundred cars and appears to be reasonably satisfactory. The earliest models have now operated about 70,000 miles in commercial use.

Inspection of Fig. 28 will show that the web of a conventional steel tire is clamped between two rubber sandwiches by two cheek plates connected to the hub of the wheel. The tire can thus move radially by deforming the rubber in shear and axially by deforming the rubber in compression. The rubber is so proportioned in the later wheels as to permit a radial deflection of about $^{3}/_{16}$ in. under a radial load of 8000 lb. Under the conditions of use with the maximum passenger load, the static load is less than 7000 lb. The earlier wheels were proportioned for $^{1}/_{8}$ in. deflection under 8000 lb; an attempt is now being made to obtain $^{1}/_{4}$ in. for new constructions.

The importance of resilience in the wheel is shown in Fig. 29. The graph was obtained experimentally. It shows the variation of vertical acceleration at the journal box with increasing resilience in the wheel. It will be observed that the greatest gain is made with comparatively small additional "springiness" but that a deflection of $^{1}/_{4}$ in. under 8000 lb does not by any means represent the limit of possibilities.

Evidently, the completely unsprung weight is reduced to that of the tire, its web, and the adjacent sandwich plates. Reduction of noise level is also achieved because the noise generated at the wheel-rail contact cannot be transmitted metallically to the hub and axle and from them by metallic paths to the remainder of the truck and the car body. Such noise energy as passes into the web must pass through the rubber sandwiches to find its way into the remainder of the structure.

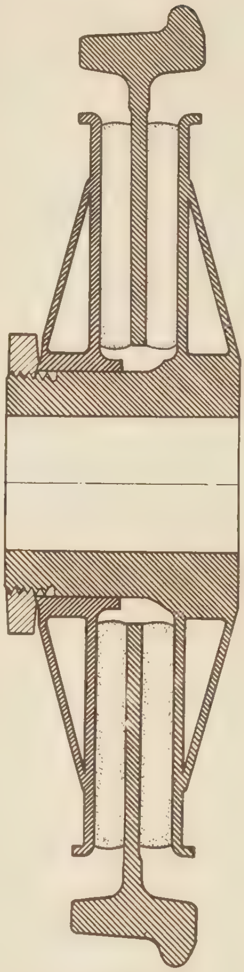


Fig. 28 Cross Section of Streetcar Wheel (Semidiagrammatic)

The noise produced by transmission of vibrations directly from the tire to the air still remains. Much effort has been expended in attempts to devise a tire construction which would transmit less in this way. A number have been found which were satisfactory from the noise standpoint but, unfortunately, none of them had sufficient strength to give the necessary life in service.

In an earlier section of this paper reference was made to the large thermal-expansion coefficient of rubber and the need for taking it into account in some cases. This wheel serves as an excellent example. The pressure put upon the rubber is determined by the position of the hub nut. The further this is screwed into the hub, the greater the pressure on the rubber. The wheels are adjusted to produce a pressure of 25,600 lb under the nut when all parts are at 70 F. The pressure under the nut then varies with temperature as indicated in Table 3. It is evident that the setting of the nut must be made with respect to the temperature existing at the time. Otherwise the wheel might be too loose at low temperatures or too tightly compressed at high temperatures.

The springs upon which the car body is carried are of a novel form shown earlier in Fig. 5. They consist of four "cylinders" of rubber bonded to flanged metal cylinders as shown in the cross section. The rubber in each section has a thick-

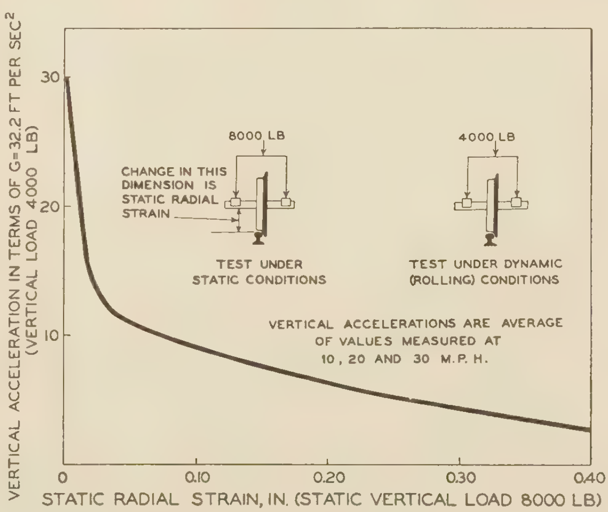


Fig. 29 Vertical Acceleration Produced at Journal Box by Impact Effect at Rail Joint (Vertical Load 4000 Lb) Vs Static Radial Strain (Vertical Load 8000 Lb)

TABLE 3

Temperature, F	Pressure under nut, lb
—20	11,000
0	14,300
+20	17,700
40	21,000
60	24,100
70	25,600
90	29,000
110	32,100

ness of about 1 in. The height of rubber diminishes from the center toward the outside so as to approach constant-shear area. This is not carried to the limit that is possible because the spring would then have inadequate lateral stability for the purpose intended. That is, if the bottom were held, the top could be moved sidewise too easily.

As constructed, the central section forms the weakest spring. The load-deflection ratio increases in successive sections as one moves outward. The umbrella structure moves downward as

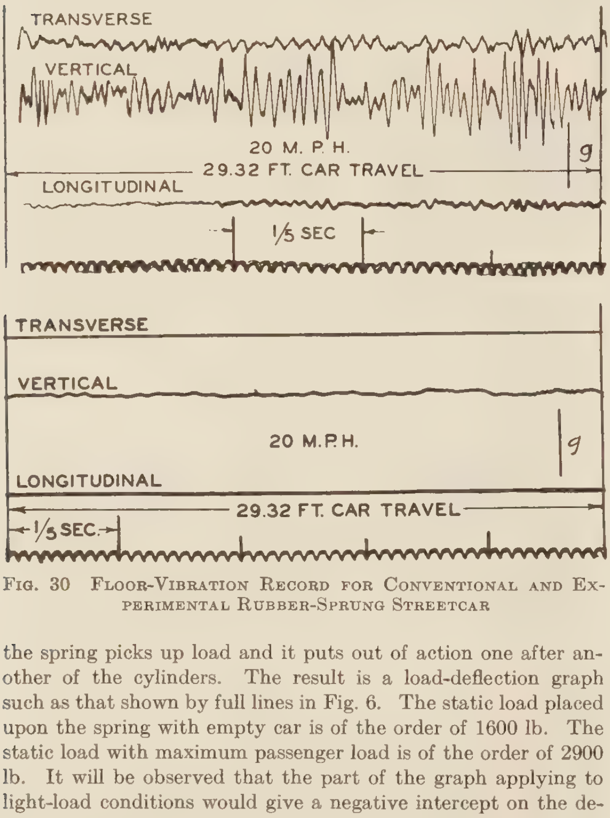


Fig. 30 Floor-Vibration Record for Conventional and Experimental Rubber-Sprung Streetcar

the spring picks up load and it puts out of action one after another of the cylinders. The result is a load-deflection graph such as that shown by full lines in Fig. 6. The static load placed upon the spring with empty car is of the order of 1600 lb. The static load with maximum passenger load is of the order of 2900 lb. It will be observed that the part of the graph applying to light-load conditions would give a negative intercept on the deflection axis if continued backward as a straight line. This makes the spring softer for such conditions. As the load increases from one limit to the other there is a step-by-step stiffening of the spring which tends to proportion it to the load in such a way as to approach a constant frequency. At the upper end the spring becomes very stiff so as to snub violent impacts.

This spring was designed to give comparatively large deflections in the vertical direction and small but measurable ones in all horizontal directions. It will be observed that the form is ideal for such purposes.

Ease of fabrication and the use of precompression are obtained by making the spring in two halves and of such proportions that they have to be pressed together to enter them into the spring containers. The central tube is entered into a receptacle at the top which serves the same purpose.

As already indicated, these springs have limited inherent damping ability for low-frequency vibrations when in use. They are therefore used with hydraulic shock absorbers in parallel with them. It has been found that on very good track the best results are obtained by setting the shock absorbers for an exceedingly small degree of damping. Stiffening of the shock absorbers produces a "hardening" of the ride.

One of the objectives of using rubber instead of steel springs was the reduction of the amplitude of the high-frequency vibrations of the floor of the conventional streetcar. This was to be achieved both by elimination of damping forces which did not attain zero values at the limits of motions and by inserting in the system a medium which would be effective in absorbing high-frequency vibrations originating below it and having to pass through it to reach the car-body floor. Another objective was the reduction of noise resulting both from passage of acoustic vibrations through the springs and from the generation of noise at the places where metallic springs are attached to the remainder of the structure.

The extent to which high-frequency vibrations can be eliminated by rubber springing of vehicles is shown by the vibration records given in Fig. 30. The record of the conventional car was obtained on good rail. It is characteristic of the extent to which high-frequency vibrations exist in the conventional streetcar. Records obtained on many different cars differed widely but in all cases they showed a great number of such vibrations of relatively great amplitude. The example given is by no means the worst obtained.

These records were made by means of an accelerometer adapted to record vibrations in the three principal directions and placed on the car floor at the center of the car. The upper record is that of transverse or crosswise vibration. The next below it shows vertical vibration and the one below that gives the fore-and-aft or longitudinal vibrations. The fourth or bottom record is a time scale. Attention is called to the small vertical line labelled g. This is the scale for vertical vibration and represents an acceleration equal to gravity. For purposes of evaluation, measurements must be made from a line of zero acceleration to positive or negative peaks.

The record for the experimental car was obtained with a rubber-sprung experimental car operating on poor rail. It speaks for itself and shows what can be done to make the streetcar a more pleasant vehicle in which to ride. It should be explained that the improvement shown is not due entirely to the use of rubber; other innovations also have had their effects. However, the rubber is the most important factor in reducing the high-frequency vibrations as shown in these records.

Inspection of Fig. 30 will indicate that the predominant frequencies shown for the conventional car are within the acoustic range. Therefore, their practical elimination in the experimental car indicates a great reduction of noise level unless other sources of noise have been accentuated. It is a fact that the new form of car is markedly quieter than the old form in spite of the fact that steel tires still run on steel rails. It is also a fact that ways are now known by which the noise level may be further reduced to a marked degree.

The Future of Rubber in Mechanical Engineering

The experience that the authors have had in adapting rubber to the springing of streetcars and to other mechanical-engineering purposes leads them to believe that there is a tremendous and hardly touched field for the use of rubber for mechanical purposes. It is a material with many unique characteristics which the mechanical engineer can use to good advantage in many applications if and when the necessary technical information becomes available. At present there are tremendous unfilled gaps in our knowledge of the properties of this material.

Inspection of A.S.T.M. Standards and Tentative Standards for rubber and methods of testing it is all that is necessary to show the state of knowledge of this material with respect to those matters which concern the engineer who would use it for mechanical purposes. References to it in mechanical-engineering textbooks and handbooks are almost nonexistent. On the other hand, the tremendous advances that have been made in the pneumatic tire within a generation show what can be done when attention is concentrated on the development of a particular type of rubber product.

If this material is to attain the position it seems to deserve in the field of mechanical engineering it must be investigated comprehensively as have, for example, steel and other commonly used metals. There is here an almost virgin field for mechanical-engineering research. It is necessary that the subject be lifted out of the field of practically pure empiricism; that rational explanations of complicated phenomena be produced.

It is possible that research will prove rubber to be subject to exactly the same laws of elastic behavior as are metals but that certain characteristics are exaggerated and others submerged in such ways as to make it appear quite different. It is also possible that a thorough study of rubber may serve to enlarge and modify our present views with respect to the elastic behavior and composition of material in such a way as to result in a more nearly perfect interpretation of the behavior of more commonly used engineering materials.

Whatever the outcome, the subject is one of tremendous potentialities and, it is believed, of relatively tremendous importance. But it must not be attacked in any superficial way if worth-while results are to be expected. Rubber is not just rubber any more than steel is just steel. Inspection of the records of tests of rubber in the literature will show that they are in many respects like those of steel made a few generations ago.

It is true that rubber technicians have made great advances in the basic study of rubber compounds in the last few years. Most of them are employed by rubber manufacturers and their findings remain the properties of their employers. However, the time has now arrived when a better general understanding of this material is necessary if it is to fill the place it seems destined to fill among engineering materials.

Processing of Rayon Staple

By HEATH O. KENNETTE,[1] NEW YORK, N. Y.

Rayon staple has been produced for a number of years, but only recently came into prominence. It produces a fabric which is distinctly different from any other textile fiber. Because of its ability to blend readily with all other known fibers, it offers almost unlimited possibilities in fabric development. At the present time, it is being used extensively in dress goods, men's suitings, necktie fabrics, upholstery, plushes, towels, knitted outerwear, and a great variety of novelty fabrics. Because of its distinctive qualities, rayon staple has earned its position in the textile industry and no doubt will become one of the most prominent textile fibers of the future.

THE manufacture of rayon staple by the producers is practically identical with that of continuous-filament rayon yarn up to the spinning machine, but because of the different spinning conditions and subsequent operations, a special plant is required. The same care and supervision is just as imperative in making rayon staple as that required for continuous-filament rayon yarn.

When regular rayon yarn is spun, 100-denier 60-filament, we have 60 separate continuous filaments forming a thread. This yarn is then purified and processed to the finished product as a separate unit. Rayon staple made by this process would be prohibitive in price; therefore the producer was forced to develop a spinning machine that would spin a yarn containing thousands of filaments as a unit. After spinning, the staple is purified, cut to length, dried, opened, conditioned, and baled. The staple, as received by the mill is in a bale, similar in size and density to that of cotton.

Rayon staple is manufactured by both the viscose and acetate processes and can be produced in several degrees of luster such as bright, semidull, and dull. The denier per filament can also be controlled to any desired size. The most common at present are 1.5, 3, or 5.5 denier per filament, $1^1/_2$ in. in length. The fiber can be cut to any length desired. The length and fineness of the fiber affect the strength of the yarn. The longer the fiber up to two inches, and the greater the number of filaments in the cross section of the spun yarn, the stronger the yarn.

The hand and character of the fabric can be changed by the use of different size filaments. The 1.5 denier per filament rayon staple produces a soft fabric. The 3 and 5.5 denier per filament produce fabrics with a different hand more closely resembling different grades of wool. By blending these fibers together and with other fibers, such as wool and silk, many desirable and unusual effects are obtained. The blending operation is usually done before the picking, but the type of yarn desired governs where the blending will be made.

Conditioning of Staple by Mill

The proper conditioning of rayon staple by the cotton mill before processing, and the proper control of relative humidity through the mill, particularly in the picking and carding operation, has been found to be most helpful. The staple should be opened, fluffed up, and allowed to normalize in a humidified area under mill conditions for at least 24 hours before running. The relative humidity at this stage and throughout the carding and spinning mill should be maintained close to 55 per cent for viscose-process staple and close to 65 per cent for acetate staple.

Picking

As rayon staple, both viscose- and acetate-processed, is supplied in a reasonable well-opened state free of foreign matter no cleaning operation is required. The main object of the picker on rayon staple is to prepare the staple for good carding by thoroughly opening the staple and producing a uniform, well-formed lap. In general, the least amount of picking employed to produce a satisfactory lap, the better will be the condition of the fiber.

The question as to whether blade or carding beater gives the best results is subject to much debate. Both types are being used successfully. Generally speaking, a carding beater will cause more neps than a blade beater, but by thoroughly breaking up the small lumps and fleecing of the staple, a more uniform lap is produced.

The speed of the beaters depends largely on local conditions and can best be answered by the mill. We have found that blade beaters operating near 1000 rpm, and carding beaters near 800 rpm give good results. Fan speeds slightly in excess of the beater speeds work satisfactorily.

Because of the greater density of rayon staple as compared to cotton, it is generally necessary to reduce the speed of the hopper feed. The production of the picker on rayon staple is comparable to that of a good grade of cotton.

The weight of the lap depends largely on the counts to be produced. A 12- to 14-oz per yard finished lap gives a good working weight.

Rayon staple can be successfully handled by the conventional cotton pickers or by the one-process picker without major changes.

Carding

It has been found in general, that if the card is in good shape and is doing satisfactory work on cotton, no trouble will be experienced in changing over to rayon staple. Normal card settings used on a good grade of cotton work equally well on staple. The top flats should be run as slowly as possible, approximately one inch per minute, and the top edge of the stripping plate should be set as close as possible so as to minimize the amount of strips. These strips normally can be reworked without trouble.

Sometimes trouble is experienced by the web's sagging or breaking. This can usually be corrected by adjusting the comb

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until October 11, 1937, for publication at a later date. Discussion received after the closing date will be returned.

and by maintaining the proper relative-humidity conditions. However, under some conditions it is necessary to increase the speed of the calender rolls.

Much has been written about removing the mote knives and replacing them with a steel undercasing, but we have not found this necessary. Normally, the finer the wire of the card clothing up to 120 or 130, the better the results. We have also found that straight-wire and metallic card clothing give very good results.

The card settings shown in Table 1 have been used successfully.

TABLE 1

Doffer speed	6 to 9 rpm
Cylinder speed	165 rpm
Licker-in speed	360 to 425 rpm
Doffer to cylinder	0.007 in.
Licker-in to cylinder	0.007 in.
Flats to cylinder	0.010 (all round)
Feed plate to licker-in	0.012 in.

Drawing Frame

It has been found that much of the trouble experienced in subsequent operations is traceable to too close a roll setting on the drawing frames. Rayon-staple fibers are easily stretched, therefore much care must be given to proper spacing of the rollers. This will be governed largely by local conditions such as the bulk of the sliver produced, weights on the rollers, type of rollers, etc. We prefer leather-top drawing rollers, but metallic rollers also give good results. Judgment must be used in adjusting the tension gear and selecting the proper-size trumpet to prevent stretching.

The following roll settings have given good results on 1.5 denier $1^1/_2$ in. staple under average cotton mill conditions: First to second, $1^7/_8$ in.; second to third, 2 in.; third to fourth, $2^1/_8$ in.

The speed, draft, and production of rayon staple on the drawing frame is comparable to that of long-staple cotton.

Slubber, Intermediate, and Roving Frames

Because of its nature, rayon staple does not require the twist that cotton does, and so the front-roller speed on the fly frames would be excessively high unless some change is made. Normally the speed of the fly frames is reduced about 25 to 30 per cent to give the same front-roller speed as used on cotton. Care must be used in selecting the lay and tension gears to build a good package without stretching the roving. Drafts normal to long-staple cotton give satisfactory results.

We have found that the following twist factors will give satisfactory results on 1.5 denier, $1^1/_2$ in. staple:

Slubber, 0.70 to 0.75 times the square root of the hank roving produced

Intermediate, 0.75 to 0.80 times the square root of the hank roving produced

Roving frame, 0.80 to 0.90 times the square root of the hank roving produced.

We have found that leather-top rollers give the best results on fly frames. Local conditions, twists, roller weights, and size of roving, govern roll settings. We have used the following roll settings satisfactorily: Slubber and intermediate, front to second and second to third, $1^{15}/_{16}$ in. and 2 in. Roving frames, front to second and second to third, $1^7/_8$ in. and $1^{15}/_{16}$ in.

Spindle speeds within the following limits have been used successfully: Slubber, 450 to 650; intermediate, 650 to 950; and roving frame, 950 to 1250 rpm.

Spinning

In the spinning of rayon staple, drafts, speeds, and production normal to cotton can be used. It has been found that rayon staple yields its maximum strength with a twist factor between 2.75 and 3.25. Sometimes it is desirable to go higher than this to get the desired hand to the fabric. Rayon staple can be twisted to produce a good crepe yarn, but as we go up in twist, the strength of the yarn comes down.

Because the back rollers on the spinning frames are nonadjustable and are set at less than the fiber length of $1^1/_2$ in., the floating middle top roller has become standard practice in spinning of rayon staple.

Cork-top spinning rollers give excellent results on crepe, coarse, and medium yarn. On fine yarn, leather-top rollers give the best results.

Generally speaking, the conventional system of drafting has given better results on medium and fine yarns; however, the long-draft system has been used successfully on coarse counts.

Spooling and Warping—Slashing

Any type of spooling and warping satisfactory for cotton can be adjusted to use rayon staple satisfactorily.

The cotton slasher is used on rayon staple principally because of the large volume of moisture to be dried out and the fact that this equipment is available in most mills. The slasher should be equipped with driven drums and temperature controls. The stretch should be held down as low as possible and should not go much over $1^1/_2$ per cent.

The question as to what type of size to use depends largely on the type of finishing the cloth will get. Good weaving results can be obtained from both gelatin- and starch-base sizes.

Weaving

Weaving of rayon staple should be considered from the standpoint of rayon rather than cotton. Rayon staple like denier yarn is subject to variation in humidity and tension, and for this reason the same precaution should be observed. If the loom is set up with this in mind, no trouble will be experienced in weaving rayon staple.

In processing rayon staple through the cotton equipment, no major changes are necessary. It has been found that rayon staple will stretch readily, therefore more care must be given to roll settings and twist factors than on cotton. Rayon staple requires a relative-humidity condition close to 55 per cent on viscose process and close to 65 per cent on the acetate process, throughout the mill for best running. If one will keep in mind the principles employed in the carding and spinning of fine long-staple cotton, the problem is not complicated. With these simple rules in mind, the average mill will be able to make a satisfactory yarn from rayon staple.

Mechanics of Synthetic-Fiber Weaving

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This paper describes the machinery that was used in the early attempts to weave rayon fabrics. It traces the course of machine development, correlating this, as far as possible, with the development of the fabrics and synthetic fibers themselves.

Changes that were made in existing looms to enable rayon to be woven are described. Comparisons between the modern rayon loom and the old types that were used for weaving rayon are given. A description of the latest appliances for weaving rayon is included, as well as a discussion of the problems that are occupying the attention of the rayon-weaving industry and textile-machinery manufacturers at present.

ACTIVE interest of loom manufacturers in synthetic-fiber weaving started in 1925. In that year, the first real attempt was made to investigate the problems of rayon weaving which, until then had been studied mainly by the mills that were endeavoring to use rayon for various purposes.

At the outset, rayon was used in the cotton-weaving industry for decoration in shirting and dress goods. It was woven with reasonable success on automatic bobbin-changing cotton looms of the type shown in Fig. 1 not only because most of the decoration was in the warp but also because the decoration that was inserted through the filling was of such heavy counts and low twist that the weaving difficulties which later were evident were not a serious obstacle in the manufacture of the goods. By 1925, the use of rayon had been extended beyond mere decoration. Bedspreads with cotton warp and rayon filling were shown at the Made-in-Carolina Exposition in September and October, 1925. Brassiere cloth of rayon and cotton was produced in a number of southern mills, and various jacquard drapery fabrics using cotton warp and rayon filling made their appearance. Several of these fabrics were produced on automatic cotton looms, many of which were of the Crompton dress-goods type. The looms were equipped with undercam harness motions, dobbies, and jacquards and with two or four cell box motions and magazines.

Because this type of loom had been built originally for the cotton industry, it was not suitable for rayon weaving, even for the fabrics mentioned. Investigation showed that the primary difficulties in the mills were

(1) Retaining the filling on the bobbin when it was in the shuttle
(2) Retaining the filling on the bobbin while it was in the magazine
(3) Obtaining satisfactory tension in the shuttle eye
(4) Bruising of yarn by the filling detector which is responsible for indicating the absence of filling yarn on a bobbin

Attempts were made to overcome these troubles by various means. To prevent the yarn from shelling or sloughing off the bobbin, the bobbin itself was sometimes treated with sizing or some kind of oil, after having been wound. At the same time, the number of turns per traverse was increased in the winding. Despite these attempts to overcome the difficulties of weaving rayon on automatic cotton looms, almost all successful weaving was done nonautomatically. In the southern mills particularly, nonautomatic cotton looms, (Fig. 2), both plain and drop box, were used. To make these looms operative, however, certain changes were made, the most important of which were

(1) The rising roll takeup was replaced by the type using a chain and friction driven cloth roll; and a 20-in. circumference take-up drum was substituted for one having a $14^1/_8$-in. circumference
(2) The side stop motion was replaced by a center filling stop motion
(3) The steel race plate was replaced with wood, sometimes covered with felt or similar fabric
(4) The vibrating warp rail was replaced by a 4-in. round wooden whip roll supported by the so-called Durkin thick and thin preventer
(5) Bobbin length was reduced from 8 in. to $7^3/_8$ in. or less
(6) Fur and other materials as well as various tension devices were incorporated in the construction of the shuttles which, in most cases, were reduced in size to decrease the shed opening
(7) A feeler was added to stop the loom when the filling was exhausted
(8) Four- or six-bank warp stop motions of the mechanical or electrical type were added

In addition to these improvements, looms were generally overhauled to eliminate play in the working parts. Brake lining was put on the brake band in an effort to stop each loom before the reed touched the fell of the cloth, and the beams and beam heads were made as true as possible.

All this activity in the cotton mills went principally into production of heavy voiles until, in 1928, a movement toward crepe manufacture was started. This fabric already had been made by one of the rayon companies which installed in its plant, as early as 1925, some 2 × 1 dobby silk looms to demonstrate the feasibility of weaving synthetic fabrics of this type. The Knowles 2 × 1 nonautomatic silk loom is typical of the silk looms that were in use in the silk mills of the country. Silk crepe was woven on it. Consequently, the logical step was for all rayon crepe to be tried on this machine. A few silk mills did enter the field, but the industry, as a whole, apparently did not adopt rayon wholeheartedly until the finer deniers of yarn began to make their appearance.

Much early work in all rayon crepe weaving was done on nonautomatic looms by cotton mills which, in several instances, converted 1 × 1 looms to 2 × 1 units by the addition of a box motion, after having made all the alterations that have been enumerated.

THE AUTOMATIC LOOM OF 1928

In 1927, Crompton & Knowles Loom Works, realizing the

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deficiencies of the looms that were being used by the cotton mills for weaving fine cotton and rayon fabrics, started work on a new cotton loom. This machine was of the automatic bobbin-changing type and was designed with many improvements to make possible automatic weaving of all-rayon fabrics that were being produced by the cotton mills. The first of these new looms went into operation in 1928. Although the majority of them were sold for weaving straight cotton fabrics, such as mar-

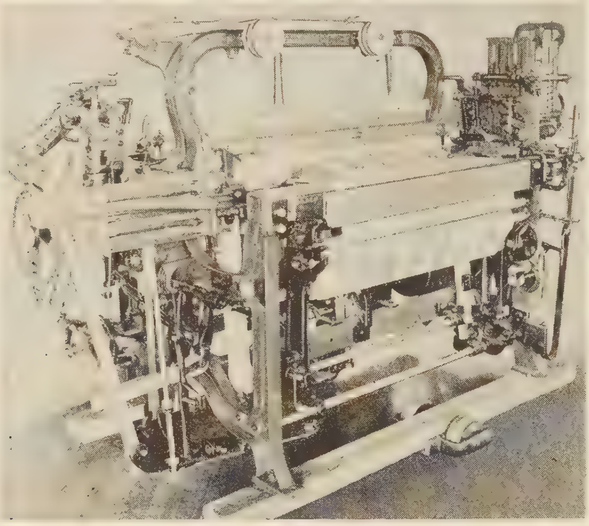

Fig. 1 Automatic Gingham Loom Used in Early Attempts to Weave Rayon-Decorated Fabrics

quisette clip-spots, handkerchiefs, shirting, and dress goods, many went into mills that were weaving rayon. Attempts to operate these automatic bobbin-changing looms on rayon soon showed that something further was necessary, particularly as regards the automatic feature. Despite the improvements, the automatic bobbin-changing mechanism still gave trouble. The principal causes were essentially the same as those that had been experienced with the old automatic bobbin-changing cotton looms that first were used for weaving fabrics with rayon decoration. To enumerate these difficulties again, they were

(1) Difficulty in retaining the filling on the bobbin while it was in the magazine
(2) Difficulty in threading the shuttle on the first pick after the transfer of the full bobbin into the shuttle
(3) Uneven tension on the filling on the first pick after transfer
(4) Drawn-in filling ends
(5) Broken filling, resulting from the transfer
(6) Cutting of filling by the filling detector or feeler

Experiences with this loom indicated need for a different type of automatic machine. Accordingly, work was started on a shuttle-changing automatic loom which would eliminate the difficulties incident to looms embodying automatic bobbin-changing. This shuttle-changing principle had been used experimentally for years on various types of looms, particularly on certain duck looms. Most of these shuttle-changing looms, however, were not a thorough success commercially, except for one type of shuttle-changing mechanism, in which the loom stopped momentarily when a new shuttle was inserted. Even that machine was not entirely satisfactory as far as weaving rayon was concerned since stopping sometimes left a mark in the cloth. Except for this defect, however, the shuttle-changing principle had the characteristics that were necessary to successful synthetic-fiber weaving. For this reason, the building of an automatic shuttle-changing loom that would transfer shuttles without stopping was undertaken.

Early Shuttle-Changing Looms for Rayon

The first installation of this automatic shuttle-changing cotton loom for rayon weaving was made in 1930. Each machine was equipped with a 20-harness dobby, a center filling stop motion, a mechanical side-slip feeler, and a take-up drum 20 in. in circumference. The shuttle size was $15 \times 1^9/_{16} \times 1^3/_{16}$ in. This installation was tested on various fabrics of all-rayon and rayon and cotton mixed. These included a flat crepe made with 64 picks per inch of 100 denier, 40-filament viscose filling and 80 ends per inch of single 46s cotton which were given a crepe twist; and various lining constructions having about 44 picks per inch of 150 denier, 24-filament viscose filling and 88 ends per inch of 150-denier 24-filament viscose warp. The new looms were compared with nonautomatic cotton looms and automatic bobbin-changing looms from the standpoint of loom stoppage and quality of cloth. They also were run at speeds as high as 216 picks per minute. All this experimental work proved the superiority of the shuttle-changing principle for rayon work and the practicality of transferring shuttles without stopping the loom.

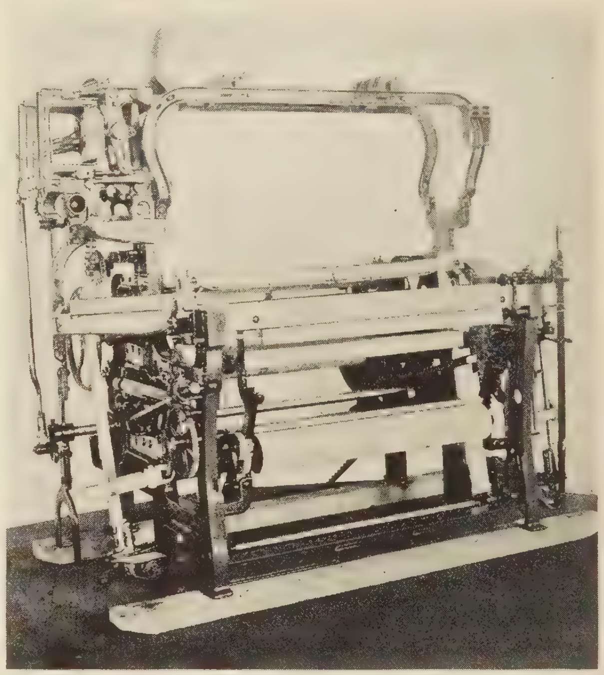

Fig. 2 Nonautomatic Cotton Loom Used for Weaving Practically All Early Rayon Fabrics

While this work was going on, another important advance was started in 1930. In response to the request of an important silk mill which had proved itself to be an outstanding leader in the use of rayon, an installation of 2 × 1 automatic shuttle-changing cotton looms was built for weaving rayon crepe. Each loom was equipped with a 20-harness dobby, a 24-in. circumference take-up roll, a center filling stop motion, and an automatic letoff. It used shuttles measuring $15 \times 1^9/_{16} \times 1^3/_{16}$ in. and had an internal electric feeler which indicated the exhaustion of the filling and put in motion the mechanism that transferred shuttles from the two-cell magazine. Experiences with this loom soon demonstrated that a machine other than a cotton loom was necessary to give the desired results. In other words, the shuttle-changing

feature was found to be generally suitable but the loom itself had many features that were not right for the weaving of fine denier yarns.

The Super Silk Loom

Coincident with the attempt to develop an automatic shuttle-changing rayon loom using the basic construction of the new cotton loom, still another important development was in process. In 1928, work on a new silk loom was started with the idea of offering to the real-silk industry a simplified machine that was capable of producing high-quality fabric at low cost.

The first trial installation of this new loom, later to be known as the Super Silk or the Type S loom, was shipped in the early part of 1930. Several months of painstaking testing and redesigning followed. The original construction was changed, not only to improve the operation of the trial installation on real silk, but also to embody certain features that had been demonstrated to be needed by mills weaving rayon. These requirements were emphasized in 1929 and 1930 by certain cotton and rayon mills which saw the need for using a silk loom and which bought either new or secondhand silk looms of the Knowles type for weaving rayon.

A nonautomatic super silk loom was shown at the 1930 Southern Textile Exposition in Greenville. In 1931, shipments were made to a number of mills. Curiously enough, however, very few of the looms went on real-silk work. The vast majority were started on rayon. The type S loom had found its place.

The looms that were shipped in the first year were mainly cone harness looms of the nonautomatic type. A few were of the automatic bobbin-changing type. Later, dobbies and Knowles heads were applied. The outstanding addition, however, was the application of the shuttle-changing mechanism which, as stated earlier in this paper, had proved itself to be essential in the automatic weaving of fine rayons. The combination of the shuttle-changing mechanism and the type S loom produced the first successful automatic loom for rayon-crepe weaving.

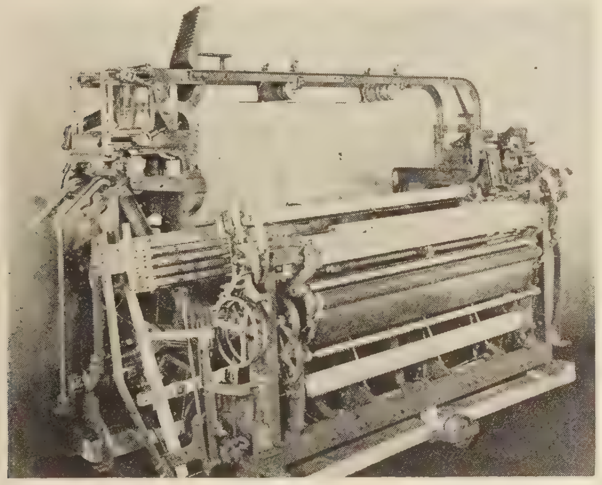

Fig. 3 Automatic Shuttle-Changing Loom With Four-Cell Shuttle Box and Two-Cell Magazine for 75 Per Cent Automatic Operation

The first of the shuttle-changing type S looms were built with electric filling detectors working in conjunction with a positive 2 × 1 box motion. Subsequently, in 1933, the mechanism was changed so that the electric feeler could operate with a call box motion of the 2 × 1 type. A later innovation of 1935 was the so-called 75 per cent automatic shuttle-changing type S loom (Fig. 3) in which two cells of a four-cell shuttle box contain shuttles that have their filling replenished automatically by the shuttle-changing mechanism, and two cells, shuttles in which the filling is replenished by the weaver. This loom was developed for fabrics having the body woven with the two automatic shuttles and the decoration with the two shuttles that are refilled manually.

Without dwelling further on the history of rayon-weaving machinery, some discussion should be presented here to show how the present equipment compares with the old looms, both cotton and silk. Certain outstanding differences between the type S and the Crompton cotton looms account for the success of the former in rayon weaving. In the first place, the warp beam is farther away from the harnesses than it is in the old cotton loom. Again, the crankshaft is positioned so that the harnesses can hang straight in the newer loom, whereas, in the cotton loom with a dobby of the same capacity, the harnesses were pulled forward at the bottom so that they might clear the crankshaft. The shed opening is smaller as the result of the smaller shuttle which is used in the type S loom. The circumference of the take-up roll is $31^1/_2$ in. as compared with $14^1/_8$ in. in the original and 20 in. in the revised Crompton cotton loom. A similar comparison between the type S and a typical silk loom, such as was used by the real silk industry and some early rayon weavers who abandoned the idea of using a cotton loom, shows that the modern rayon loom, in its over-all dimensions, is more like the old-time silk loom than the old-time cotton loom.

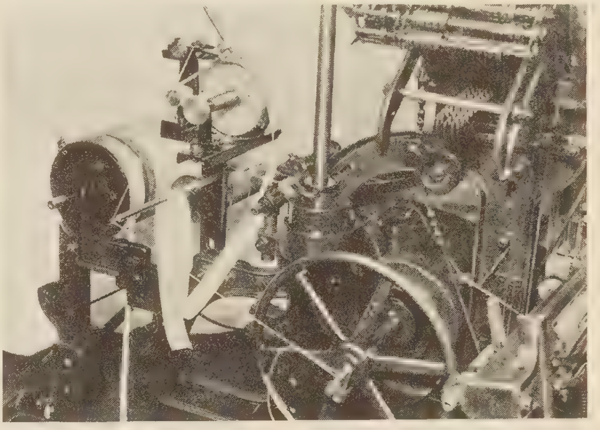

Fig. 4 Apparatus Used to Test the Characteristics of Loom Letoff Mechanisms

Important Loom Mechanisms

While the main dimensions of a weaving machine naturally have an intimate bearing upon the results that can be obtained with it, the mechanical details of design and construction are the items that, in the last analysis, determine its success or failure. For this reason, a few of the more important elements of construction in the type S loom will be discussed. The mechanisms that really constitute the backbone of a weaving machine are (*a*) the letoff, (*b*) the shedding mechanism, (*c*) the lay, (*d*) the picking mechanism, (*e*) the take-up, and (*f*) the magazine and its related parts. Many other small parts of the machine and auxiliary items of equipment contribute to the satisfactory operation of the whole; but these groups of mechanism are of primary importance.

The Letoff. The letoff mechanisms used in the weaving industry can be classified under three general types; friction, whip-roll control, and dead weight.

The first type is the most common. Examples can be found

on cotton and silk looms. Ordinarily, in the cotton industry, the friction member that encircles the friction head consists of steel chain, whereas, in the silk industry, it consists of a piece of braided rope. Although this type of letoff is used in weaving rayon, it is not perfect. Tests that have been conducted by special recording apparatus, Fig. 4, indicate that the warp letoff at each pick is not uniform.

The automatic letoff which operates on the whip-roll control principle also has its drawbacks. Here, as the warp tension builds up, the whip roll is displaced, thus allowing the throw of the letoff driving mechanism to be increased. In other words,

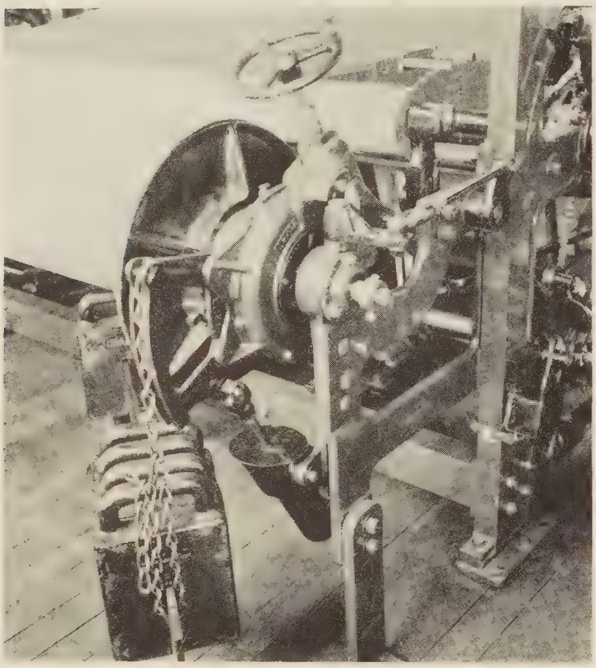

Fig. 5 Automatic Letoff of the Dead-Weight Type

the apparatus can be likened to the action of a thermostat in regulating the temperature of a room. While reasonably sensitive, its action has a definite lag, depending upon the frictional resistance of the moving parts and the lost motion in the connections between the whip roll and the drive. Some letoffs of this type are worse than others. The best of them shows sufficient lack of uniformity in the amount let off to be unsatisfactory on fine rayon work, even though it has its place in the manufacture of cotton goods and coarse rayon.

Letoff mechanisms employing the dead-weight principle have been found by experiment and practical use to be superior to the friction or the whip-roll control types. In this type of mechanism, the tension on the warp is determined by the weights that are hanging on the letoff, as shown in Fig. 5. The same results could be obtained if a rope were fastened to the beam head so that it would pass around the periphery and support a weight hanging on the free end of the rope. With this arrangement, the rope would be wound on the beam head as the warp was pulled forward by the take-up. The function of the moving parts in the dead-weight type of letoff is to compensate for the tendency of the rope to wind on the head. The weight is let back with reference to the beam as the beam rotates forward, thus maintaining a given position of the weight with reference to the rest of the loom.

Among the studies that have been made on letoffs is one in which the tension variation during one revolution of the crankshaft of the loom was determined. A curve of warp tension in terms of degrees of crankshaft rotation was plotted. This chart was made from readings taken by slow-motion pictures. It indicates an appreciable increase in the warp tension as the harnesses open, as well as at the time of beat-up. Obviously, tension with the harnesses open is a detriment because, at that time, the warp ends are supported on the thin edges of the heddles. The endeavor, then, is to eliminate, as far as possible, the increasing tension, as the harnesses open, by the dead-weight principle. With that type of mechanism, the beam can roll forward as the shed is formed, thus keeping the tension in the warp approximately constant. Without this relief, the tension varies as is indicated in Fig. 6. Some variation occurs with the dead-weight letoff, because of the inertia of the beam and the weights. The exact amount has not been determined because slow-motion pictures of the mechanism were not taken until recently. The fact remains, however, that practical results with the dead-weight letoff indicate a condition of uniformity which is superior to the characteristics of the other two types of letoff.

Shedding Mechanism. In the modern rayon loom, the shedding mechanism is an undercam-harness motion, a dobby, or a Knowles head. Some looms have been built with a cone harness motion and some with a jacquard, but the great majority are of the three types mentioned. These mechanisms have characteristics that differ somewhat. The undercam harness motion and the Knowles head have a definite dwell, whereas the dobby because of its crank motion, gives a continuous movement to the harnesses, except on one pick when the dobby hooks are not on the dobby knives. A comparison of these mechanisms, as plotted in Fig. 7, shows that the Knowles head changes the harnesses in 130 crankshaft deg and allows them to dwell in the open position for 230 deg. Similarly, the cam motion changes the harnesses in 210 crankshaft deg and allows them to dwell for 150 deg. In some instances, particularly on five-harness work, the dwell is reduced to 120 deg to make the shape of the cam less abrupt.

The effect of the dwell is to give less interference between the shuttle and the shed lines than is the case where a dobby is used. To have shedding conditions equivalent to those obtained with a cam-harness motion or a Knowles-head motion, the harnesses must be opened wider with a dobby. This naturally places an increased strain on the warp. On multiharness work, the Knowles head has another advantage over a dobby in that it can be equipped with sectional cylinder gears. These can be adjusted so that all the harnesses do not cross at the same time. This characteristic is particularly valuable on heavy sley work where the warp ends interfere with each other when the harnesses cross at the same time as they do with a dobby.

Although this comparison shows that the Knowles head motion possesses points of superiority as compared with a dobby, it should not be construed as an indication that more heads than dobbies are used in rayon weaving. Of course, many dobby looms are giving good, practical results. The fact remains, however, that the nearest approach to the ideal shedding condition can be obtained with the Knowles head.

The Lay. The lay motion of the modern rayon loom includes one mechanism which is essential to the elimination of marks, a swing-reed release motion. This mechanism consists of a series of fingers that holds the backstay in position when the loom picks and that yields when it beats the filling into place. The purpose of the mechanism is to give the fell of the cloth a blow of uniform intensity regardless of its position. If the weaver makes a pick-out and lets the cloth back a little too much, the swing reed yields sufficiently to compensate for the error. Until this motion was put on the type S looms, no satisfactory means for eliminating start-up marks was available. Reeds on old cotton and silk looms were held in what amounted to a fixed position in most

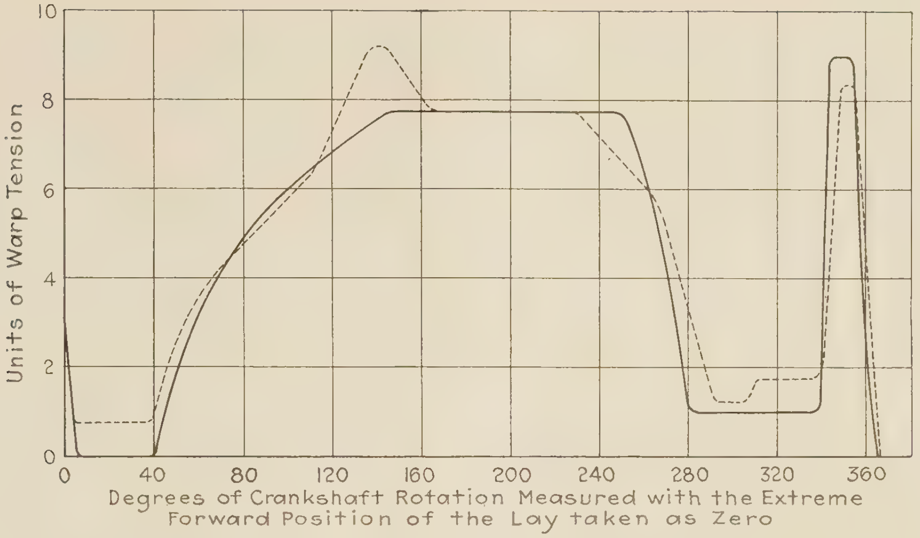

Fig. 6 Variation of Warp Tension for Two Complete Revolutions of the Crankshaft of a Dobby Super Silk Loom

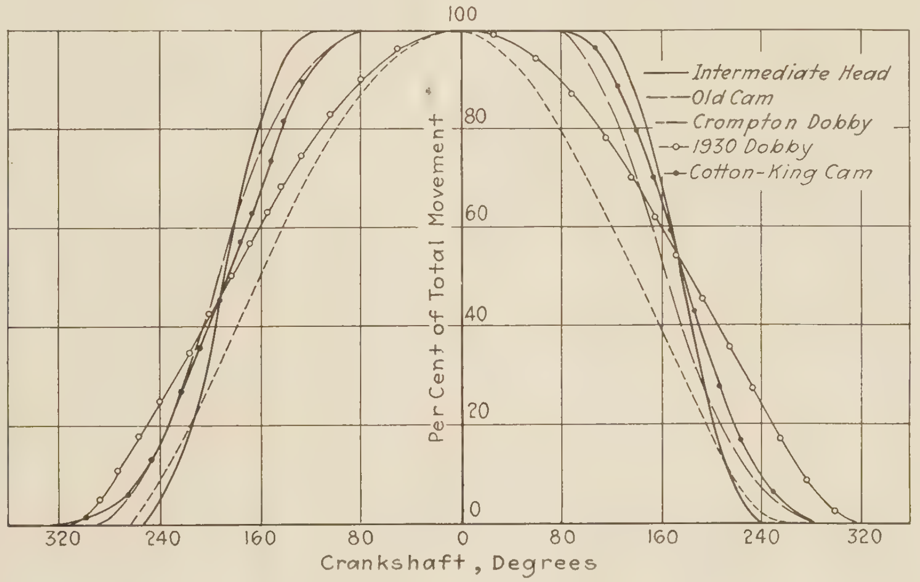

Fig. 7 Comparison of Harness Motions

instances, with the result that to start a loom without leaving a mark in the cloth was virtually impossible.

Incorporated in the backstay that supports the reed is a glass rod that prevents the warp from dropping on the race plate if the harnesses are set too low in the loom. Modern rayon looms not only have this feature but also are arranged so that the reed can be adjusted to line with the shuttle boxes.

In the looms now being built, a shuttle box with a wide mouth is used. This arrangement, together with the corner of the lay beam and the race-plate covering, has a bearing on successful operation. The wide-mouth shuttle box was adopted to prevent cutting of the filling by the shuttle as it entered the box. The corner of the lay beam is made of wood and is covered with corduroy where formerly a brass casting was used. A wooden race plate also is covered with corduroy where, in the older looms, the race plate was of uncovered steel or wood in the cotton looms and wood covered with billiard cloth in the silk looms.

In the conventional shuttle-changing lay, provision is made for a shuttle either 15 in. long × $1^9/_{16}$ in. wide × $1^3/_{16}$ in. high or 15 in. long × $1^3/_8$ in. wide × $1^1/_{16}$ in. high. These shuttles take paper tubes which generally are $6^1/_2$ in. long and are wound to a diameter of approximately $^{15}/_{16}$ in. for the large and $^3/_4$ in. for the small shuttle. These diameters could be somewhat larger but, in most instances, are kept small enough to permit the use of fur or sheepskin as shuttle lining.

In the lay of the Crompton automatic cotton loom that was used for rayon weaving by some cotton mills when they started producing synthetic fabrics, a shuttle $15^3/_4$ in. long × $1^7/_8$ in. wide × $1^3/_8$ in. on the back wall × $1^5/_{16}$ in. on the front wall was used. This shuttle took a bobbin 8 in. long × $1^1/_8$ in. wound diameter. Usually the bobbin was of the wooden type such as is used in an automatic bobbin-changing loom. These facts are brought out to show how much smaller than the original cotton shuttles are the standard rayon shuttles used on the type S shut-

tle-changing loom. Between these extremes are many variations which have been used on nonautomatic looms by rayon-weaving plants.

The most recent development in bobbins and shuttles is the so-called flat-cop shuttle. Various designs are in use but, in general, the dimensions are approximately $16^1/_2$ in. long $\times$ $1^{15}/_{16}$ in. wide $\times$ $1^1/_4$ in. on the back wall $\times$ $1^1/_{16}$ in. on the front wall. These shuttles take an elliptical bobbin or cop which averages about $1^7/_{32}$ in. on the major axis and about $^{15}/_{16}$ in. on the minor axis when wound. The bobbin itself is generally about $7^3/_4$ in. long. No opinion concerning the adoption of the flat-cop arrangement by the rayon industry as a whole can be ventured as yet. Mills are divided in their ideas, some finding it workable, others, an impossibility. Those who argue for it point out that,

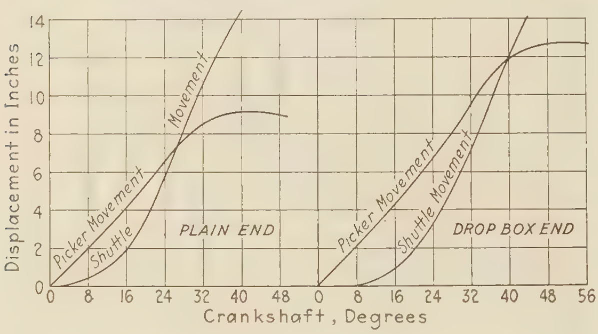

Fig. 8 Comparison of Picker and Shuttle Movement

as compared with the conventional shuttle-changing package of the larger size already mentioned, the oval bobbin will take approximately twice as much yarn and that, because of the shuttle's peculiar shape, the shed opening need be no greater than with the larger of the two shuttle-changing types. While this, of course, is true, the cross section of the oval package is not alone responsible for the increased carrying capacity. Approximately 50 per cent of the increase arises from the cross section and the other 50 per cent from the increased length of the package.

Picking Mechanism. Before describing the work that has been done, one comment with reference to picking motions perhaps should be made. Frequently, the question is asked, "Why hasn't somebody invented a better method for propelling the shuttle?" The answer is that, despite all the mechanisms which have been built, nothing has been found which is better from a practical standpoint than the cam-operated mechanism that always has been used. Some excellent electrical means for propelling a shuttle and others employing mechanical or pneumatic devices are in existence. They all have their drawbacks, however, from a cost and operating standpoint. For these reasons, the loom builders have striven to improve the conventional mechanism.

In the old looms, picking cams were made by cut-and-try methods. Now, however, a theoretical displacement curve for the picker is worked out, taking into consideration the speed and width of the loom, the time and distance that are available for bringing the shuttle up to speed, and the time that can be allotted to the passage of the shuttle across the loom. From this displacement curve, the shape of the cam on its pitch line is determined. A steel template of this shape is made and put in a grinding machine which generates the working surface of the cam in a pattern. The pattern is then used in the foundry to produce the chilled-iron casting that constitutes the picking cam itself.

A few interesting facts have been determined through slow-motion pictures that were taken to aid in the design of picking cams. The actual shuttle velocity of a 54-in. silk loom was 40.9 fps even though the cam was originally laid out to give 25.4 fps. This increased velocity results from the resilience of the picker stick which must be taken into consideration when the picking cams are designed, not only because of its contribution to the velocity but also because of its effect upon loom timing.

The distance that the picker sticks bend in actual operation is rather surprising. Slow-motion pictures show that, as the result of this bending and whatever stretching is in the picker-stick connections, the shuttle is as much as 2 in. behind the position in which it should be just after the loom starts to pick. For instance, in the silk loom that was tested, the shuttle, during 12 deg of rotation of the crankshaft, should have moved 3 in. from its initial position. Actually, as shown in Fig. 8, it moved only 1 in., showing that the picker stick was bent out of line by approximately 2 in.

Another interesting result of the moving-picture analysis was the determination of the shuttle-velocity drop as the shuttle moves across the loom. Taking the maximum velocity of the shuttle as it leaves the shuttle box as 100 per cent, the velocity as it enters the selvage was found to be 87.6 per cent. At the center of the loom, it was 66.6 per cent and, at the opposite selvage, it was 59.5 per cent. In other words, the shuttle loses approximately one third of its velocity in traveling from selvage to selvage.

Take-Up. When compared with the cotton-loom mechanisms that were used for rayon, the take-up motion of the modern rayon loom has as its outstanding characteristic the size of the drum, $31^1/_2$ in. in circumference, as compared with $14^1/_8$ and 20 in. in the older looms. Another essential difference between the two looms is in the cloth-winding and guide rolls. In the latest type, the cloth comes over a wooden strip or a glass rod on the breastbeam; passes over a guide roll, around the take-up drum, around a pressure roll, and over another guide roll; and then goes to the cloth roll which is driven by a chain and friction arrangement from the drum. In the cotton looms, on the other hand, the cloth passed over the breastbeam which was made of wood instead of steel, then over a steel guide roll, around the take-up drum, over another guide roll, and finally to the cloth roll which was held in contact with the drum by spring-supported bearings. Thus, the cloth was wound on the cloth roll by the frictional contact of the cloth roll with the cloth on the take-up drum, a situation which introduced a considerable rubbing of the cloth on itself. This arrangement was soon found to be inadequate and was changed to one employing a lower winding roll driven by a chain which was not unlike that now employed in the modern rayon loom.

In the older looms, the take-up was not fitted with pressure rolls as are the rayon-weaving looms. Some of these new looms, in fact, have two pressure rolls, one on the front and one on the back of the take-up drum. The front one is covered with thick Rosella felt and the other with a harder and denser felt that is similar to billiard cloth. In the cotton industry, coverings usually consist of various grades of perforated tin and sandpaper. The latter also has been used in the real-silk industry. With the advent of rayon, however, these coverings were found to be too harsh; consequently, crepe rubber was adopted. This, however, showed deterioration and, subsequently, was replaced with a material called Texalox and various cork and rubber compositions. More recently, a composition of cork and Duprene has been used to eliminate the difficulties with rubber.

Take-up gearing has been the subject of study, not only from the standpoint of tooth profile but also from that of the number of teeth in contact, the face width of the gears, and the adequacy of the supporting bearings, as well as their lubrication. Where the gears of the old looms, in many instances, had cast teeth, they now are made with cut teeth. Tooth profile has been improved to eliminate the involute interference that exists on ordinary

pinions and gears which are cut with the regular standard tooth length and a $14^1/_2$-deg pressure angle. Thus, with improved bearings, the gears can be adjusted and maintained with the minimum backlash.

In cotton looms, usually no provision is made for lifting the hold-pawl when the loom is stopped. In starting the loom, the weaver usually gains a pick and leaves a mark on rayon. The Type S rayon loom is arranged to lift the hold-pawl so that the loom will not take up a pick when the weaver starts the loom, after having stopped it for any reason. This item is a refinement which, although seemingly inconsequential, actually makes a great difference in the quality of the product.

Magazine. As far as the shuttle-changing magazine, Fig. 9, and its operation are concerned, not much can be said without a long explanation of details. The essential point with regard to this mechanism is that the shuttles, as they lie in the magazine, are threaded so that, when they are transferred into operation, no difficulty is experienced with tension on the first pick or with threading the shuttle eye. Similarly, the question of drawn-in ends of filling, which is present in most bobbin-changing mechanisms is eliminated.

Before leaving the construction of modern rayon looms, another point should be emphasized. Prior to 1928, cotton and silk looms were somewhat like agricultural machinery. The parts were not thoroughly machined and were fitted to the main framework by hand. Now, however, with accurately machined parts, looms can be built on an assembly line as are other modern products, like the machine tools and automobiles. The result is a solidly and accurately built mechanism which is capable of fine adjustment and which can operate at high speed and efficiency to produce first-grade cloth with minimum effort on the part of the employee.

Auxiliary Equipment. Outstanding among the investigations of auxiliary equipment that is essential to weaving-equipment operation is one concerning warp beams. In the cotton industry, warp beams, until recently, were made with a wooden barrel, into which gudgeons were driven at each end. The heads were fastened to the barrel by rods or bolts and were made with the flange cast integral with the friction drum. No attempt was made to have the flanges adjustable. In the silk industry, the beams usually had a wooden barrel with a through shaft, and friction heads, 10 in. in diameter, that were fastened at each end by long rods running from one beam head to the other. With these beams, paper was used so that the warp might be spread to the desired width on the beam. The scheme was to run a certain number of yards on the beam. Then, a paper covering was inserted, and the operation continued for a few yards. This process was repeated until a beam of the desired yardage was made.

In view of the small yardage and the time involved in handling the paper, consideration was given to making a beam that would eliminate papers and would permit the beaming of greater yardage. The first paperless warp beam was sent to a rayon mill in 1929. This beam had 16-in. adjustable flanges that could be moved lengthwise of the beam to any desired width on a threaded sleeve. It also had a wooden barrel, and a through shaft. Since 1929, thousands of these warp beams have gone into the rayon industry. Subsequent developments, however, have increased the flange diameters to the point where the beam journals have been increased from $^7/_8$ in. in diameter to $1^1/_8$ in. to avoid bending the journals. Developments in steel beams also have been made so that, at present, all-steel paperless warp beams are available. These beams were designed to increase the strength without increasing the weight.

One of the difficulties encountered in beam design is the tremendous end pressure against the flanges which is produced in beaming a rayon warp. Apparatus for determining this end pressure has been built, but, to date, the work has not gone far enough to permit making generalizations beyond the observation that rayon warp acts like a pile of sand. Unsized warp is like dry sand, and sized warp resembles sand mixed with clay. As would be expected, therefore, end pressure on the flanges is greatest with the unsized warp. In testing, unsized warp was found to have an end pressure three times that obtained with sized warp.

The development of paper tubes for filling and of fiber-covered shuttles, the substitution of steel heddles for cotton harnesses, the use of silk reeds instead of pitch band reeds, and improvements in temples and temple rolls all have contributed to more perfect weaving. Similarly, the use of oil guards around the dobby, take-up gears, and picking motion has minimized the hazard of cloth spoiled by oil spots.

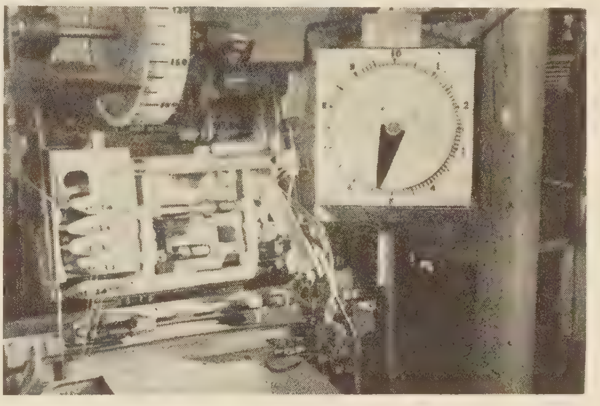

Fig. 9 Shuttle-Changing Magazine of the Type S Loom and Recording Apparatus Used in Slow-Motion Picture Analysis

Economic Status of Rayon-Weaving Equipment

In concluding this paper, some reference to the equipment of the silk and rayon industry, as a whole, should be made. Approximately 73,000 box looms are suitable for this work. This total includes no cotton looms. Of the looms 76 per cent are 10 or more years old; 50 per cent, 15 or more years old; and 26 per cent, 20 or more years old. Only 12 per cent are of the latest type S shuttle-changing model to which this paper has been devoted. All these machines have been built since 1931, and practically all are devoted to rayon weaving.

These figures show that the equipment available for rayon weaving, at least as far as box looms are concerned, is still far from modern, from the standpoints of both age and ability to weave automatically. Thus, improvements that are being made by the machinery manufacturers should be paralleled by modernization of the rayon-weaving plants. The great strides that have been made in less than ten years are a credit to the industry as a whole, but they constitute only a good beginning for what history will describe as one of the greatest contributions to the welfare and enjoyment of the civilized world.

Discussion

The Use of Alloy Steels for Side Frames and Bolsters of Freight-Car Trucks[1]

H. W. Stertzbach.[2] The author mentions the efforts made to meet the present high test requirements with minimum weight of side frames and bolsters made in the present carbon or grade-*B* steel. In these efforts the writer believes that the manufacturers who have earnestly devoted their talents and labors toward this achievement in grade-*B* steel have been rewarded by a good measure of success, as is evidenced by the fact that the present types of side frames of the integral-box type in grade-*B* steel are considerably lighter than the earlier types. For example, the frame for the 50-ton car has been reduced from around 650 lb to about 590 lb, a weight saving of nearly 10 per cent. While this weight saving was in itself worth something to the railroads, the more important phase of this progress in improvement of grade-*B* designs is the greater strength developed in the lighter designs—a situation which was quite naturally and properly taken advantage of by the railroads in stepping up the strength requirements accordingly.

The actual improvement in strength, as determined by static tests, shows the lighter designs at least 15 to 20 per cent stronger than the early designs.

This was accomplished, of course, by improved foundry and metallurgical practices as well as by skill and invention, or if you please, engineering, in the economic distribution of metal. Other important tools in this work are the empirical methods of stress calculation which afford a fair degree of reliability for comparative purposes and the more laborious but much more accurate methods of calculation, of which the celluloid-model method is perhaps the most satisfactory. Static and fatigue tests had an important part in all of this work, but the final and most important proof has been the performance in service of the modern grade-*B* designs, in defense of which nothing need be said here.

The writer believes that the reappearance of the truck with separate journal boxes is to be expected only in special cases, because of the attendant increased first cost and weight. With reference to the possibility of spring mounting the side frames when separate journal boxes are used, this would demand a change in the almost universal practice of supporting the brake parts of the freight-car truck on a member not subject to spring travel unless some kind of auxiliary support is devised to be supported by the journal boxes.

That part of the paper under discussion which refers to The Association of American Railways having adopted tentative dynamic-test specifications in the development of lightweight alloy side frames should be understood to mean that the dynamic test is not to become a part of the specification covering routine acceptance of truck side frames. Such a specification would necessarily lead to rather arbitrary interpretations of test results. The interpretation of the dynamic-test results in the present development of truck side frames of alloy steels is in the hands of an A.A.R. committee of engineers who have been selected on the basis of their ability to properly weigh the test results with judgment founded on the comparative performance of groups of tests rather than that of a single specimen. Furthermore, fatigue testing is a long and expensive procedure, several days being required for a single test. Due to the time required, such acceptance testing would lead to confusion in scheduling production and would naturally result in delays in delivery. Therefore, the proper sphere for dynamic testing is laboratory study and development where progress of fatigue is interpreted with judgment and with proper regard being given to an established background of comprehensive testing, manufacturing, and service experience.

All of what is said here with reference to the proper sphere of fatigue testing is fully recognized by the A.A.R. Car Construction Committee as is shown by the following expression in their circular DV-836 dated May 26, 1935, and with reference to the development of high-tensile freight-car truck side frames.

"The present A.A.R. side-frame specifications purposely were formulated so as to provide for acceptance under static tests only when using grade-*B* steel, but the background for these test requirements with this material had previously been established through a prolonged series of dynamic tests and research.

"Consistent with present testing procedure for grade-*B* frames, dynamic-test requirements are not to be incorporated in the specifications for high-tensile steel frames."

The advantage of the use of alloy-steel truck frames and bolsters of course resides in the saving of weight, the economics of which is yet to be determined.

In so far as the saving in weight of a complete truck is concerned, the use of lightweight frames and bolsters affects a relatively small proportion of the complete weight of the truck for the reason that about 70 per cent of the weight of present trucks, with grade-*B* steel castings, is made up of wheels, axles, and brake parts.

TABLE 1 EXTENT OF THE USE OF ALLOY-CAST-STEEL TRUCK SIDE FRAMES AND BOLSTERS AS APPLIED TO FREIGHT CARS IN THE UNITED STATES THROUGH 1936

Date placed in service	No. of cars	Type of car	Axle capacity, journal size, in.	Name of owner	Alloy-steel Frames	Alloy-steel Bolster
1935	25	Refrigerator	5×9	Northern Refrigerator Co.	Yes	Yes
1935	15	Hopper	$5^1/_2 \times 10$	C. & O. R. R.	Yes	Yes
1934	100	Hopper	$5^1/_2 \times 10$	B. & L. E. R. R.	Yes	Yes
1934	5	Hopper	$5^1/_2 \times 10$	P. & L. E. R. R.	Yes	Yes
1934	5	Hopper	$5^1/_2 \times 10$	C. B. & Q. R. R.	Yes	Yes
1936	1000	Hopper	6×11	B. & L. E. R. R.	No	Yes
1936	1000	Hopper	$6^1/_2 \times 12$	B. & L. E. R. R.	No	Yes
1936	1000	Box	$5^1/_2 \times 10$	Union Pacific R. R.	No	Yes
1935	1	Hopper	$5^1/_2 \times 10$	Rustless Iron Corp.	Yes	Yes
1934	1	Hopper	$5^1/_2 \times 10$	Pressed Steel Car Company	Yes	Yes
1935	1	Box	$5^1/_2 \times 10$	Pullman-Standard Car Mfg. Co.	Yes	Yes
1935	1	Box	$5^1/_2 \times 10$	Mt. Vernon Car Co.	Yes	Yes

Alloy-steel castings such as side frames and bolsters subjected to high stresses present the necessity of all the care and control of manufacture that even the most experienced foundry can command, as will be realized by consideration of the fact that when sections are decreased additional adequate safeguards must be provided against the possible effects of unavoidable imperfections in material and variations in thickness. Special care in heat-treatment and the extra capital investment involved in the production of alloy castings are also important economic factors. Castings in this grade of steel are required to have the letters

[1]Published as paper RR-58-3, by Donald S. Barrows, in the November, 1936, issue of the A.S.M.E. Transactions.

[2]Chief Mechanical Engineer, The Buckeye Steel Castings Company, Columbus, Ohio. Mem. A.S.M.E.

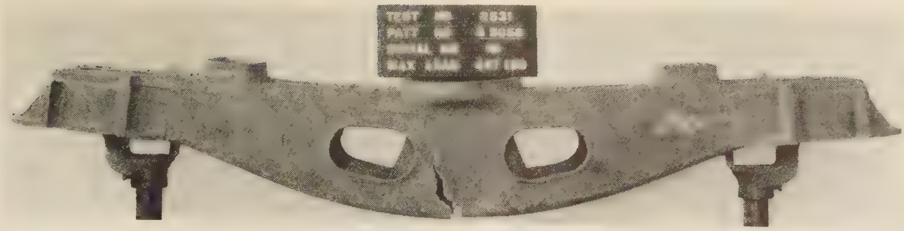

Fig. 1 One of the 50-Ton Capacity Bolsters After Applying Maximum Load

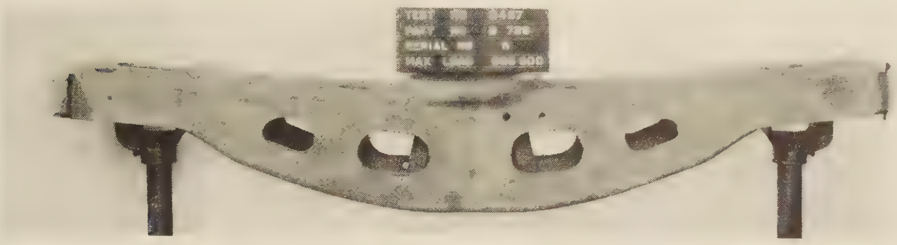

Fig. 2 One of the 70-Ton Capacity Bolsters After Applying Maximum Load

Fig. 3 One of the 90-Ton Capacity High-Tensile Lightweight Bolsters After Applying Maximum Load

H.T. cast on them to serve as a signal to the repair man that proper facilities for preheating and heat-treatment should be at hand if repairs are to be made. In fact, present A.A.R. instructions prohibit all welding on alloy truck frames and bolsters pending further developments and experience.

Obviously the factors directed toward increased costs of lightweight designs are numerous, and present costs indicate that the additional and necessary expense may or may not be justified.

Table 1 of this discussion shows the extent of the use of alloy-cast-steel side frames and bolsters as applied to freight cars now in service in the United States. Briefly, this list shows that after the completion of cars now under construction, there will be 3154 cars in service with lightweight alloy truck bolsters, but only 154 of these cars are equipped with lightweight side frames. Of the 3154 car sets of lightweight bolsters, three fourths of these were furnished by one manufacturer, namely, The Buckeye Steel Castings Company.

Any or all of the manufacturers who are cooperating with the railroads in the development of these lightweight designs are

TABLE 2 RESULTS OF STATIC TESTS OF 50-TON CAPACITY ALLOY CAST-STEEL TRUCK BOLSTERS

	Average results of two tests	A. A. R. requirements
Weight, lb	644[a]	......
Transverse test:		
Deflection at 82,000 lb, in	0.08600	0.120 max
Set after 154,000 lb, in	0.00225	0.025 max
Vertical test at side bearing:		
Deflection at 120,500 lb, in	0.05400	0.080 max
Set after 192,500 lb, in	0.00200	0.025 max
Vertical test at center plate:		
Deflection at 120,500 lb, in	0.09000	0.120 max
Set after 231,000 lb, in	0.00200	0.025 max
Elastic limit, lb	327,500	....
Ultimate load, lb	482,500	423,500 min

[a] Have side-bearing pockets cast integral which weigh approximately 22 lb per bolster.

TABLE 3 RESULTS OF STATIC TESTS OF 70-TON CAPACITY ALLOY CAST-STEEL TRUCK BOLSTERS

	Average results of two tests	A. A. R. requirements
Weight, lb	800	... ...
Transverse test:		
Deflection at 101,000 lb, in	0.0730	0.120 max
Set after 192,000 lb, in	0.0030	0.025 max
Vertical test at side bearings:		
Deflection at 149,000 lb, in	0.0520	0.080 max
Set after 240,000 lb, in	0.0025	0.025 max
Vertical test at center plate:		
Deflection at 149,000 lb, in	0.0860	0.120 max
Set after 288,000 lb, in	0.0020	0.025 max
Elastic limit, lb	420,000	....
Ultimate load, lb	629,420	528,000 min

TABLE 4 COMPARATIVE RESULTS OF STATIC TESTS OF 90-TON CAPACITY TRUCK BOLSTERS OF ALLOY AND GRADE B STEEL

	Alloy steel		Grade-B steel	
	Average results of two tests	A. A. R. requirements	Average results of two tests	A. A. R. requirements
Weight, lb	882	....	1192	....
Transverse test:				
Deflection at 120,000 lb, in	0.0810	0.120 max	0.0460	0.075 max
Set after 230,000 lb, in	0.0035	0.025 max	0.0060	0.025 max
Vertical test at side bearing:				
Deflection at 177,500 lb, in	0.0580	0.080 max	0.0405	0.055 max
Set after 287,000 lb, in	0.0040	0.025 max	0.0080	0.025 max
Vertical test at center plate:				
Deflection at 177,500 lb, in	0.0920	0.120 max	0.0660	0.075 max
Set after 345,000 lb, in	0.0035	0.025 max	0.0080	0.025 max
Elastic limit, lb	453,000	....	395,000	....
Ultimate load, lb	675,375	632,500 min	689,825	632,500 min

ready to put more of these castings in service, and with the present predominance of lightweight bolsters, it would appear very desirable to equip perhaps a limited number of cars with more of the lightweight side frames.

This development of lightweight steel castings for freight cars is not confined to side frames and bolsters. It may be of interest to add here that quite a number of cars are also equipped with lightweight couplers, draft yokes, striking castings, and body-bolster center braces.

Tables 2, 3, and 4 of this discussion giving results of tests of 50-ton, 70-ton, and 90-ton alloy-cast-steel truck bolsters are presented to supplement the results tabulated by the author. Figs. 1, 2, and 3 of this discussion show a 50-ton, 70-ton, and 90-ton alloy-cast-steel bolster, respectively, after testing.

F. G. Lister.[3] The interesting and timely data furnished in this paper is worthy of careful consideration. The freight-car truck has been the subject of a great deal of thought for a good many years, and more so in the last few years since the railroads have speeded up their trains, requiring more attention to the bolsters, truck frames, and brake rigging than ever before. The service is more severe. Heavier and more frequent braking has thrown more strain on the bolsters and side frames.

The truck frame, in its evolution from the old arch bar to the present U-section cast-steel frame, has undergone many changes, and not until the dynamic-testing machines were developed by the manufacturers of steel frames and research work commenced in connection with the design of the frames was it possible to distribute the metal through the structure to uniformly control the stresses encountered in actual road service. However, this meant a cast-steel side frame of heavy sections, making the extra weight undesirable. Because of this fact consideration has been given to the use of alloy steels.

It is admitted that a worth-while saving in weight at no sacrifice of strength can be made by use of alloy steels in both truck side frames and bolsters. Alloy steels of high strength and dependability have been available for many years, but these have been too expensive to justify their use in freight-car construction. The problem resolves itself into the selection of an alloy steel which has high strength, ductility, and good casting properties and which at the same time is reasonable in cost. Fortunately, the last two years have seen the development of a number of new alloy steels which meet these requirements with a considerable degree of success. In these steels, high physical properties are obtained through use of the cheaper alloying elements such as silicon, copper, and manganese, with or without the addition of small amounts of chromium or nickel. When the experimental work now under way has been completed, several of these now low-priced alloy steels should find a useful place in car castings for the reduction of dead weight, and this should be especially true for truck bolsters and truck side frames.

Author's Closure

Mr. Stertzbach may be correct in believing that the reappearance of the separate journal box is to be expected only in special cases, but the steady increase in operating speeds of freight equipment has forced more careful consideration of the efficiency of journal-box lids and dust guards. The day has passed when it can be reasonably held that a dustproof lid joint may be made between a pressed-steel lid and a rough cast box face or that oil can be retained and water kept out by means of a simple basswood dust guard floating in a cored dust well. It avails little if an improved dust guard makes a water- and oil-tight seal around the axle, if there is no equally tight seal between the guard itself and a machined surface at the rear end of the box. The machining of the front and back ends of a separate journal box should be less expensive than the corresponding machining of boxes cast integral with a side frame, and this is the basis for the author's original statement. With separate boxes there would be the further advantage of angular flexibility in a horizontal plane between journal box and frame, thereby avoiding damaging contacts between journal bearings and the inside of the box.

As to the spring mounting of side frames, Mr. Stertzbach evidently had in mind provision for full spring travel over the boxes, and the elimination of the spring-mounted bolster as was done in the "Verona" truck. The author was not considering the elimination of the bolster springs, but their retention of present travel with the addition of short-travel springs over the boxes in order that the frames themselves would be protected against fatigue and the individual wheels made more responsive to track irregularities as a very definite protection against derailments in high-speed service.

The author is in agreement with Mr. Stertzbach that the requirement of a fatigue test for each order of side frames would be unworkable, but the present tentative requirements of the A.A.R. for nonstandard lightweight frames provide for a fatigue or dynamic initial test on four frames and a similar test after the production of the first one thousand frames. The specification later states: "As soon as a sufficient background of dynamic- and static-test experience has been gained with lightweight side frames of various designs and materials to warrant future acceptance on the basis of static tests only, ... no further dynamic tests will be required, except in the case of frames of substantially different design or material which have not previously passed a dynamic test."

Table 1 of Mr. Stertzbach's discussion specifically refers to alloy frames and bolsters applied to domestic service in 1936. In addition, the Gould Coupler Corporation furnished lightweight alloy frames and bolsters for four hundred Brazilian cars.

[3] Superintendent of motive power, St. Louis-San Francisco Railway, Springfield, Mo. Mem. A.S.M.E.

Square-Edged Inlet and Discharge Orifices for Measuring Air Volumes in the Testing of Fans and Blowers[1]

R. E. Sprenkle.[2] In his recommendations to the Power Test Committee No. 10 on Centrifugal and Turbo-Compressors and Blowers of the A.S.M.E. of the use of thin-plate orifices for testing fans and blowers in place of the more expensive flow nozzle or the more elaborate procedure of making pitot-tube traverses, as was originally intended, the author has performed a valuable service to industry. Not only has he verified the splendid work initiated by Ebaugh and Whitfield,[3] but he has extended this work to cover the discharge orifice as well. As a result of this work,[1] the author has found the orifice to be just as accurate as the flow nozzle, and that a test can be made much faster than when using a pitot tube. Therefore, there is no justification for the use of any other device than the orifice, especially when it is apparent that the cost of a thin-plate orifice for ducts as large as are required for fan testing is but a fraction of that for the same size nozzle.

It is perhaps unfortunate that the physical aspects of the

[1] Published as paper AER-58-7, by Lionel S. Marks, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Mechanical Engineer, Bailey Meter Company, Cleveland, Ohio. Mem. A.S.M.E.

[3] "The Intake Orifice and a Proposed Method of Testing Exhaust Fans," by N. C. Ebaugh and R. Whitfield, Trans. A.S.M.E., vol. 56, December, 1934, paper PTC-56-3, pp. 903–912.

orifice as well as the data obtained should have been compared only with the V.D.I. orifice and its coefficients in this paper, when there exist today unexcelled data on the American orifice. The writer refers not only to the many excellent papers previously given before this Society on characteristics of the American types of orifices, but also to the A.G.A.–A.S.M.E. Orifice Committee report.[4] In this report are data which are even more complete than the V.D.I. data in so far as practical application is concerned. It is true these data do not

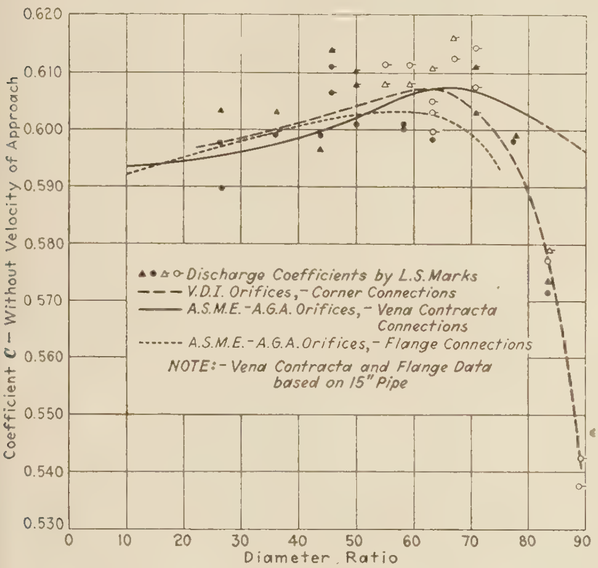

Fig. 1

cover the discharge or the inlet orifice, but on the other hand, neither do the V.D.I. data. As the author has stated, the V.D.I. specifications for discharge orifices are exactly the same as for duct orifices. Therefore, the writer believes that a comparison of the author's studies with the latest A.G.A.–A.S.M.E. data[4] will be of considerable value in ascertaining if orifices as prescribed by the joint A.G.A.–A.S.M.E. Committee can be used successfully as discharge orifices.

First of all in considering this comparison it should be understood that while the author states that his orifices are of a modified V.D.I. type, they are in reality the same as our American orifices using flange connections as recommended by the American Gas Association, and which are included in the report.[4] In fact, the author's orifices conform as closely to the American as to the V.D.I. pattern which incidentally the writer believes is a fortuitous circumstance.

It is important to state at this time that the best way of plotting orifice coefficients is to show the discharge coefficients alone without including the velocity of approach, also to use diameter ratio instead of area ratio. By so doing the real characteristic of the device is shown, that is, it reveals whether the actual discharge coefficient is stable with increasing or decreasing diameters ratios, or whether it changes rapidly and thus becomes unstable. On the other hand, the inclusion of the approach-velocity factor with the discharge coefficient covers up such trends in such a way that they are not readily discernible. It is important to know the magnitude of such changes in coefficient value as it is inadvisable to use an orifice having an unstable coefficient if perfect dependence is to be placed upon its results.

Curves showing coefficients of discharge for A.S.M.E. orifices using vena-contracta pressure connections, for A.G.A. orifices using flange connections, and for the V.D.I. rifices using corner connections, all without the approach-velocit factors, are shown in Fig. 1 of this discussion. As is well-known, vena-contracta connections are placed one diameter preceding and in the plane of the smallest jet area following the orifice, whereas the flange connections are placed 1 in. preceding and following the orifice. The location of the V.D.I. corner connections is very clearly illustrated in Figs. 2 and 3 of the paper. The curves in Fig. 1 of this discussion show that the characteristics of the three types of orifices are practically the same up to a diameter ratio of 60 per cent, and that there is little to choose from in favor of either type up to that point. However, the curves begin to separate quite widely beyond the 60 per cent diameter-ratio point, and it will be noted that the coefficients of both the A.G.A. orifice with flange connections and of the V.D.I. orifice with corner connections begin to drop quite rapidly. This is one reason why the A.G.A.–A.S.M.E. report[4] has not recommended the use of flange connections for orifices of about 75 per cent diameter ratio.

In contrast to the sharp drop of the coefficient curves for both the V.D.I. and the flange-connection types, the curve for the vena-contracta orifice is comparatively flat, with about the same slope for diameter-ratios above and below 70 per cent. There is no reason, therefore, why orifices using vena-contracta connections cannot be used with certainty up to diameter ratios of 0.825. In fact, vena-contracta orifices of such maximum sizes are used extensively in commercial practice, and with very satisfactory results.

The author's data, taken from Fig. 4 of the paper and corrected to the basis of discharge coefficients alone, are plotted in Fig. 1 of this discussion. The magnitude of the spread between individual points may suggest the scale used in this plotting is too large. However, this is the same scale used in making the preliminary analysis of the A.G.A.–A.S.M.E. vena-contracta data referred to previously (the final scale used in presenting these data was $2^1/_2$ times larger than this).

If each one of the author's points could be given equal weight, it would appear that the discharge-coefficient curve for discharge orifices could be about 0.3 per cent above the V.D.I. curve between the diameter ratio of 45 to 70 per cent. This would still come within a ± 1.5 per cent tolerance. In fact, the use of either kind of pressure connection, namely, corner (V.D.I.), flange (A.G.A.) or vena-contracta (A.S.M.E.), would still place all of the author's data within this tolerance. The writer is firmly of the opinion, however, that the use of the vena-contracta connections would enable this tolerance to be reduced to at least ± 1 per cent, and would allow the use of a larger range of diameter-ratio orifices with a smaller chance of involving an unknown error due to the rapid change in the characteristic value.

Ronald B. Smith.[5] The author has concerned himself with a potentially important problem in the field of low-pressure flow. In both this country and in Europe there have been demands, within the last three years, for test-code recommendations on thin-plate inlet and exit orifice installations. In the United States the response has been deferred; in Europe the problem has been met by increasing the tolerance on the available tests. Whatever the method of attack an atonement for ignorance appears to be important in the beginning. A tolerance increasing with the area ratio at least for discharge orifices seems rational, for if the scattering of tests at the low ratios as indicated in Fig. 4 of the paper is the result, as the author suggests, of distorted

[4] Report of the American Gas Association and A.S.M.E. Joint Orifice Coefficient Committee, The American Society of Mechanical Engineers, 29 West 39th Street, New York. N. Y., 1935.

[5] Turbine Engineering Department, Westinghouse Electric & Manufacturing Company, South Philadelphia, Pa. Jun. A.S.M.E.

inlet-velocity profile, the effects will be more serious as large area ratios are used. The pressure differentials are more frequently smaller with high area ratios, which also enhances the possible error. At present it seems probable that an orifice in a pipe or at the end of a pipe can never be used with as low a tolerance for fan or reciprocating-blower tests as it may for water or steam measurements unless effective damping can be secured.

The problem of orifice-plate thickness is a question of importance. It should not vibrate, and it should be sufficiently stiff so that it does not deflect. Even with small pressure difference when plates are 60 in. in diameter and only $^1/_{16}$ in. thick, the deflection may not be insignificant. In Europe this question has been viewed more seriously than here.[6,7] Would it not be better to specify a minimum plate thickness and cylindrical-orifice length as a function of the pipe diameter rather than to suggest the use of $^1/_{16}$- or $^3/_{32}$-in. plates without an upper limit on duct size?

In this connection the European experiments[8] indicate that the length of the cylindrical edge and the plate thickness affect the discharge coefficient oppositely. The data apply to corner taps which are essentially those employed by the author. It has been found that if the cylindrical length were about one third of the plate thickness, the effects are compensating for area ratios as high as 0.7. When the plate is less than $0.04D$, where D is the pipe diameter, the orifice length and the plate thickness may be identical without noticeable effect on the coefficient. The author's experiments seem to fall within this range, but with smaller ducts, unless a geometric ratio is chosen, they may be subject to error.

The decision to measure the differential pressures with pipe taps located 1 in. upstream and downstream from the orifice plate is a noteworthy break from European convention. The use of pressure chambers in large pipes is not only costly but unwieldy. If taps are located symmetrically around the periphery of the pipe and the pressure readings compared, it has frequently been possible to determine irregularities in the approaching stream and to correct them before testing. This is a distinct advantage for fan-test work. Rather than an arbitrary location of the taps at 1 in. upstream and downstream, the writer favors geometrically similar installation requirements with the dimensions as functions of a linear variable, say the pipe diameter. For the size tested by the author it is apparent that the choice of 1 in. lies within $0.01D$ and $0.03D$ from the pressure plate. Since this is essentially the tolerance allowed by the V.D.I., the author's comparison with the European standards should agree. Had the 1 in. dimensions been employed for ducts of 16 in. diameter, and for large area ratios, the impact pressure rise that occurs near the corner[9] would hardly have begun and differences in the coefficient of as great as 2 per cent may actually be found if this 1-in. location be adhered to. The upstream location at 1 in. is satisfactory for ducts larger than 30 in. diameter, but in smaller sizes it is sufficiently far away from the orifice so that the pressure reading will be affected by the impact built up in a variable manner depending on the area ratio. Would it not be better to accept either the corner tap or the location $1D$ upstream, and thus be rid of the difficulties?

After a study of the pressure distribution downstream of the orifice,[3] it would seem that the agreement between the author's studies and Stach's[10] on inlet orifices is more fortuitous than rational. Between a corner tap and a $0.4D$ downstream tap there may be as much as 3 per cent difference in the pressure, or possibly 1.5 per cent in the discharge coefficient. In mentioning Stach's tests it may be well to point out several possible errors. Stach's measurements are relative, inasmuch as they were made by calibrating the flow through the inlet and exit orifices with other nozzles installed in a standard manner in the line. However, the distance between the in-line meters was 18 pipe diameters which is not sufficient to be certain of damping the discharge eddies and thus preventing the discharge of one meter affecting the entrance to another. In addition the nozzles, some as small as 4 in., were made from sand castings with an uncertain amount of machining, and finally the area ratios for some nozzles were larger than 0.45 which is a region that has since been found in error[7] by as much as 0.5 per cent.

Although the author makes no mention of the use of the discharge nozzle, it may be well to point out a difficulty apart from the cost that makes the orifice more suitable. At the end of a line, most frequently when discharging a fluid or gas into one of different viscosity there is, particularly for low-pressure flows, a tendency for the flow to leave the throat of the nozzle and to form an appreciable contraction. This appears to be the result of a pressure which may be slightly less than atmospheric at the point where the nozzle profile changes from a curve to a straight throat, and where the flow temporarily leaves the wall. If noticed, this transient difficulty may be overcome by temporarily disturbing the outlet flow with a plate or by working a wire around the nozzle throat, but the uncertainty of its presence is a distinct disadvantage to the use of a discharge nozzle. A somewhat similar difficulty has been found by the writer, even when the nozzle is discharging fluids of the same viscosity (air into air) if the exit pipe is only 4 or 5 diameters long. Since the pressure within the pipe may be less than atmospheric, on account of the downstream build-up, outside air may break in and thus disturb the downstream pressure reading. The difficulty may be overcome when discharging to air by using a long exit pipe or by installing a downstream valve.

G. L. Tuve.[11] The recognition of square-edged orifices as one of the code methods for use in fan testing would indeed be a welcome simplification, and the author has done a real service in obtaining data to support his statement that "the square-edged orifice is a very reliable device for measuring air."

After using thin-plate orifices in several hundred fan tests over a period of 15 years, the writer is thoroughly convinced of their practical value. For low duct velocities the pitot tube is often of little use, and the orifice is a welcome alternative; in any test work it is certainly desirable to have two methods of measurement available for occasional checking. Much test work is done with a false sense of security regarding the accuracy of results, and it is interesting to note that in order to come within a tolerance of 1.5 per cent on pitot-tube measurements, the author states that the micromanometer was necessary.

The author departs somewhat from American practice in plotting the orifice discharge coefficients with approach factor included and by plotting against orifice area ratios instead of against orifice diameter ratios. American engineers also prefer to use a more limited range of orifice sizes because of certain disadvantages of the very large or very small ratio sizes. Again, the pressure connections used in this country are not usually those

6 "Calibration of an Orifice," by H. W. Swift, *Philosophical Magazine*, series 7, vol. 8, no. 51, October, 1929, pp. 409–435.

7 "Orifice Discharge Coefficients for Viscous Liquids," by G. L. Tuve and R. E. Sprenkle, *Instruments*, vol. 6, November, 1933, p. 201.

8 "Neuere Mengenstrommessungen zur Normung von Düsen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 5A, 1934, p. 205.

9 "Die Strömung durch Düsen und Blenden," by R. Witte, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 2A, 1931, pp. 245–291.

10 "Die Beiwerte von Normdüsen und Normblenden im Einlauf und Auslauf," by E. Stach, *Zeit. V.D.I.*, vol. 78, 1934, pp. 187–189.

11 Professor of Heat-Power Engineering, Case School of Applied Science, Cleveland, Ohio. Mem. A.S.M.E.

specified by the German standards. A wealth of data on coefficients published by or with the cooperation of the A.S.M.E. Fluid Meters Committee, has been apparently overlooked by the author. While these data refer largely to pipe orifices, the V.D.I. coefficients were also obtained on pipe or duct orifices.

Fortunately, in the diameter-ratio range of 0.3 to 0.75 (0.09 to 0.56 area-ratio range), the coefficients from the Orifice Committee report,[4] from the V.D.I., or from the A.G.A.–A.S.M.E. cooperative project at Ohio State University, all agree within less than 1 per cent. Hence, it makes little difference which is used. Moreover, the data from these sources indicate that either vena contracta, flange, or corner taps may be used without exceeding the tolerance of 1.5 per cent.

Tests have been made in the laboratories of the Case School of Applied Science, Cleveland, Ohio, in which both pipe and discharge orifices have been compared against the pitot tube, venturi meter, and heat balance (heating coil in the air stream). In addition to confirming the findings of the author, one additional conclusion is worth mentioning, that is, when an orifice coefficient deviates for some reason from standard, it is almost always high rather than low, as evidenced also by the 18 points above, and only eight points below, the curve in Fig. 4 of the paper. From this standpoint, it would seem more logical to set the tolerance limits as + 2.5 per cent and —0.5 per cent, rather than ± 1.5 per cent.

Author's Closure

Two of the discussers of this paper express some surprise that the author has not compared his results with those reported by the A.G.A.–A.S.M.E. Orifice Coefficient Committee[4] but has chosen instead to compare with the coefficients contained in the V.D.I. rules. No other procedure was actually practicable, since the American coefficients apply only to pipe or duct orifices. It is true that the V.D.I. rules give identical coefficients for discharge orifices and duct orifices and that no experimental justification for this identity is offered. The experimental work on discharge orifices used by the Germans appears to have been that of Stach[10] which gave discharge coefficients for a discharge orifice differing from the V.D.I. coefficients for duct orifices. In the V.D.I. rules, discharge orifices are given a tolerance larger than the tolerance proposed for duct orifices; this procedure may be a confession of uncertainty as to the true values of the discharge-orifice coefficients or may be an attempt to include the Stach coefficients up to an orifice area ratio of about 0.6. The fact that the V.D.I. rules give identical coefficients for the discharge orifice and the duct orifice does not appear to the author to justify the assumption that these two coefficients are in fact identical and consequently does not appear to him to justify an assumption that the values of the American coefficients for duct orifices can be considered to apply to discharge orifices.

Another point on which two of the discussers agree is in expressing a preference for the use of pressure measurements taken at the vena contracta. The author fails to perceive any application of this to the case of a discharge orifice and thinks that the discussers must have had in mind duct orifices or possibly inlet orifices. The solid-line curve of Fig. 1 of this discussion has no applicability to the author's tests.

There is also criticism of the use by the author of orifice area ratio (as used also by the Germans) rather than the orifice diameter ratio. The use of orifice area ratio appears to be preferable since it spreads out the curve in the region of higher values of the ratio which is also the region of most rapid variation of the discharge coefficient.

Mr. Sprenkle suggests that the best way to plot orifice coefficients is to show the discharge coefficients alone without including the velocity of approach. The author is entirely in agreement with him on this point and has incorporated values of the coefficient, so defined, in Table 1 of the paper. The plottings, it is true, are on the basis of the coefficient which includes the velocity of approach and this was done in order to make a direct comparison with the German coefficients. These coefficients have been adopted by the International Standards Association and it seems to the author desirable to conform as far as possible in the method of presentation of results. It has, moreover, the minor advantage of simplifying calculations for discharge.

Mr. Smith comments on the lack of definiteness in the recommendations contained in the paper for orifice plate thickness. What he has to say about the influence of the length of the cylinder edge and of the plate thickness is correct and is taken care of in the V.D.I. rules. For the range of duct diameters which the author had in mind, on which he had carried out investigations, and which are of interest in fan and blower tests, the two elements to which he calls attention become unimportant. Any moderate variation from the dimensions suggested by the author would have an entirely negligible influence on the discharge coefficient.

The status of inlet orifice coefficients is rather curious. The results obtained by Stach, by Ebaugh and Whitfield, and by the author all agree in giving an identical discharge coefficient. This consistency is puzzling because the pressure differentials on which they are based are different, Stach having used a corner tap while the other two investigations used pressure taps at a distance of 0.4 times the duct diameter downstream from the orifice. A difference in the pressure differentials measured at these two places is certain; its value is indicated in the paper of Ebaugh and Whitfield, and is stated by Mr. Smith. That the author's results agree with those of Ebaugh and Whitfield is not an inconsistency, since the same location of the pressure tap was used in both cases. The reasons for the agreement with Stach may possibly be those suggested by Mr. Smith.

Professor Tuve calls attention to the scattering of values of the discharge coefficients and to the fact that the coefficients are generally high as compared with the V.D.I. values. The discharge coefficients were obtained by comparison with volumes determined from pitot-tube traverses, the latter having previously been compared with calibrated-nozzle discharge measurements. It is the author's experience that pitot-tube traverses in fan ducts have a marked tendency to give values which are too high. This may result from swirling motion of the air, from pulsations, and from the pattern of the velocity distribution. As the calculated values of the orifice discharge coefficients reflect directly any errors in the pitot-traverse measurements, the author is inclined to regard the scattering of points shown in Fig. 1 of this discussion as being due primarily to errors in the pitot-traverse readings and not to variations in the orifice discharge coefficients. Unfortunately this belief is not susceptible of proof.

Undercooling in Steam Nozzles[1]

Charles H. Coogan, Jr.[2] Under a grant from the faculty research committee of the University of Pennsylvania, the writer is studying the phenomena which occur in the steam-jet air pump. The apparatus used by the writer consists mainly of a framework in which nozzle and sliding diffuser plates of various shapes may be placed. These various metal plates are placed between two pieces of 10 × $29^1/_2$-in. glass which are parallel and $^{13}/_{16}$ in. apart. The customary search tube is so

[1] Published as paper FSP-58-6, by J. T. Rettaliata, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Instructor in Mechanical Engineering, University of Pennsylvania, Philadelphia, Pa. Jun. A.S.M.E.

arranged that the pressure may be recorded at any point in the central plane of the two-dimensional flow.

In studying the flow of steam in the nozzles used by the writer the point of condensation in the flow of superheated steam was observed by carefully determined pressure measurements. This point of condensation was determined by the method suggested by Professor Keenan in his discussion of Professor Yellott's paper,[3] i.e., observing the point at which the pressure drop becomes zero momentarily or rises slightly. Professor Keenan plotted a curve of initial pressure against indicated pressure rise at the point of condensation. His curve indicates that the pressure rise varies directly with the initial pressure. The writer has found that for the low initial pressure of 12.5 lb per sq in. abs the pressure rise indicated was approximately 0.4 lb per sq in.

Fig. 1 of this discussion shows the pressure-drop curve for one nozzle used by the writer. It is quite obvious where the point of condensation occurs from the discontinuity in the pressure drop.

The writer has found that for the smooth nozzle, condensation

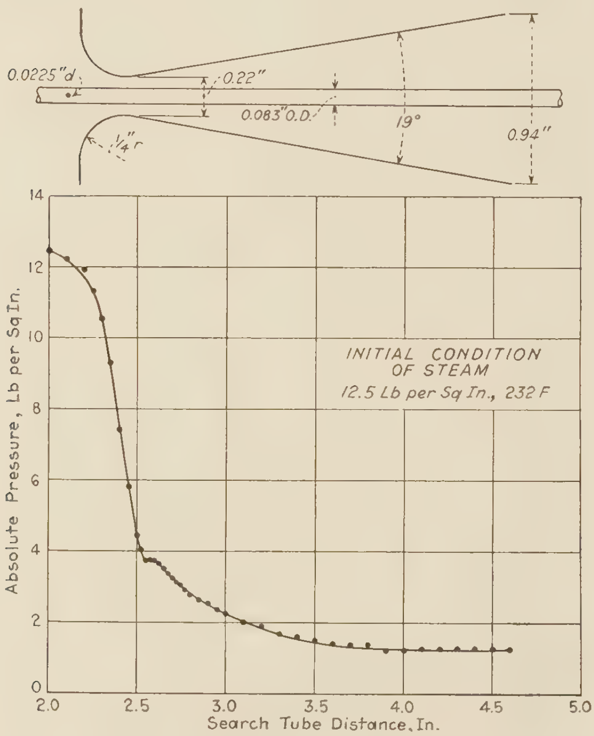

Fig. 1 Pressure Drop in a Short Nozzle

as determined by the previously mentioned method occurred at the moisture line of 4.3 per cent on the Mollier diagram. One should not forget, however, that this is based on an assumed constant nozzle efficiency of 96 per cent up to the point of condensation. Should this assumption be incorrect the location of the point would be altered.

In regard to the time required for condensation to start after the pressure has fallen below the saturation pressure, it should be pointed out that not only does the roughness of the nozzle have an appreciable effect on this time, but the length and contour have a considerably greater effect. The velocity is dependent on the

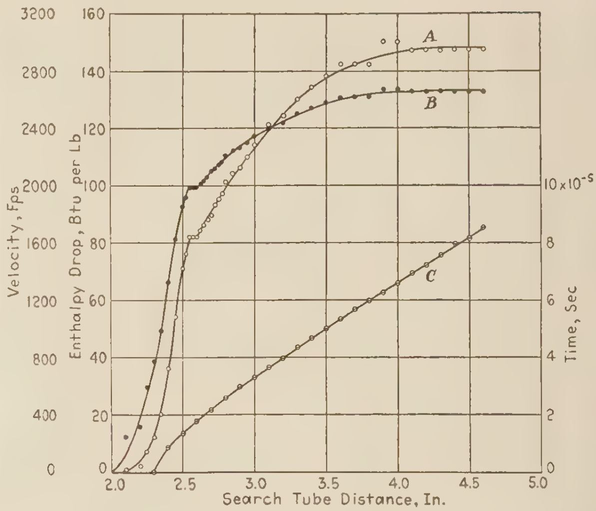

Fig. 2 Enthalpy Drop, Velocity at Various Points, and Time for Steam to Reach These Points in a Short Nozzle as Obtained by Graphical Integration

(Curve A shows enthalpy drop assuming a nozzle efficiency of 100 per cent as determined from the Mollier diagram from measured search-tube pressures. Curve B shows the velocity at various points with an assumed nozzle efficiency of 96 per cent constant throughout the nozzle and with supersaturation loss neglected. Curve C shows the total time required from 2.3 in. to each point as determined from $t = \int_{s=2.3}^{s=s} ds/V$.)

pressure drop, but the distance in which this drop occurs varies with different length nozzles. The shorter the nozzle for the same drop, the shorter the time in which condensation occurs. Condensation time for the small nozzle shown in Fig. 1 of this discussion required from 1.2×10^{-5} to 1.5×10^{-5} sec as determined by graphical integration. Condensation time, therefore, is not dependent on roughness alone.

In Equation [1] of the paper, which the author later averages to find the average velocity between the saturation line and the condensation point, the following should be noted: First, the nozzle efficiency has been assumed constant which, although not exactly so, will nevertheless give a very close approximation since the difference between the square roots of the efficiencies is smaller than the difference of the efficiencies. Second, the average velocity which the author has found is the average of velocity in the equation $v = f(H)$ where v = velocity and H = enthalpy drop. The enthalpy drop is dependent on the pressure drop in any nozzle and therefore this method of averaging involves an assumption which may not be true. The average of velocity should be found with respect to time. While the method of determining this time interval may give approximate results over a small enthalpy drop it nevertheless would give considerable discrepancies over larger enthalpy drops.

In Fig. 2 of this discussion the writer has plotted the total time required for a particle of steam to pass from 2.3 in. to any point up to 4.6 in. From this curve it is noted that the total time as determined by graphical integration is 8.5×10^{-5} sec, whereas when calculated by the method used by the author the time should have been 10.8×10^{-5} sec (neglecting supersaturation loss). The method of graphical integration to find the space-time diagram can be easily applied to cases with an assumed varying nozzle efficiency and to those in which the supersaturation loss is also considered.

There is a minimum time in which condensation will occur after passing the saturation line. The writer wonders whether the author has tested any smaller nozzles to determine the lower limit of time required.

[3] Discussion by J. H. Keenan on "Supersaturated Steam," by J. I. Yellott, Trans. A.S.M.E., vol. 56, 1934, paper FSP-56-7, p. 427.

In so far as determining the actual phenomena occurring during condensation, the writer is of the opinion that a study of the molecular structure of a gas, vapor, and liquid might be extremely helpful.

Howard Emmons.[4] The writer wishes to make a few remarks concerning the size of the droplets that exist at the point of initial condensation. The drop radii were computed by the author from the von Helmholtz equation for the variation of the vapor pressure of a liquid with the variation in the radius of curvature of the liquid surface with which the vapor is in equilibrium. The use of this equation introduces two assumptions: (1) The drops are in equilibrium with the vapor surrounding them; and (2) the drops are large compared to molecular dimensions. This latter assumption enters because of the thermodynamic character of the equation, and will be appreciated if one tries to define "surface tension" for a drop consisting of only a few molecules.

Let us consider these two assumptions. The equilibrium state to which the von Helmholtz equation applies is one of unstable equilibrium since smaller drops decrease in size while larger drops grow. As pointed out by Professor Keenan in his discussion[3] of Yellott's paper, supersaturated steam without droplets is in a state of metastable equilibrium. As soon as drops appear which are larger than the equilibrium radius they will grow to visible (i.e., light-scattering) size. Therefore, the first assumption seems reasonable for the drops upon which condensation first starts. The visible droplets may be considerably larger than those computed by the von Helmholtz equation.

Now let us turn to the second assumption, calling to mind some facts from the kinetic theory of gases. The approximate diameter of a water molecule is 4×10^{-8} cm or a radius of 2×10^{-8} cm. This is seen to be approximately one third of the size of the drops computed from the experimental supersaturation ratio. This brings us immediately to the question: How many molecules must be considered for the probability results given by a thermodynamic formula to be correct to a given accuracy? As far as the writer knows, this question has not been answered and so long as it is not answered we will have to be careful when using thermodynamic equations on the droplets of small size.

In the remainder of this discussion, assumptions (1) and (2) previously mentioned by the writer will be taken as true. Since the computed droplets contain approximately 3^3 or 27 molecules, the kinetic theory of gases provides a possible mechanism for the occurrence of these condensation nuclei. On the average each water molecule makes approximately 10^{10} collisions per second. The author has computed that the time required for the steam to pass from the saturation line to the Wilson line is approximately 5×10^{-5} sec. Each water molecule will therefore have made 5×10^5 collisions. In many of these collisions more than two molecules will be involved. In some cases the 27 or more molecules necessary to form a nucleus would collide and never again be able to break up, but instead would grow by absorbing other molecules.

It is interesting also to push assumptions (1) and (2) to the limit and note that the minimum possible drop size is equal to the radius of a molecule or 2×10^{-8} cm. Now by means of the von Helmholtz equation (Equation [13] of Yellott's paper)[5] we can compute the supersaturation ratio. By extrapolating the superheat data into the wet region, this ratio will locate a limiting position of the Wilson line. As this extrapolation must be made for a considerable distance the result is inaccurate but approximately the Wilson line for steam, as limited by the size of the molecules, lies near the 11 per cent moisture line.

A. Egli.[6] The writer would like to know why the author chose the particular steam conditions for the tests with nozzle A, represented by Figs. 2, 3, 4, and 5 of the paper. Why did he hold his back pressure so high, a condition which is not of great importance in steam turbines working with nozzles of the Laval type?

In what respect are the two following sets of steam conditions for the tests of nozzle B supposed to be "entirely different?" Condition No. 1 was an inlet pressure of 55 lb per sq in., a steam temperature of 305 F, and a back pressure of 10 lb per sq in. Condition No. 2 was an inlet pressure of 44.7 lb per sq in., a steam temperature of 280 F, and a back pressure of 8.3 lb per sq in. The ratios of back pressure to inlet pressure of the two cases are 0.182 and 0.186, respectively, that is, practically identical. Also the nozzle operation (for instance, the pressure distribution of the efficiency) would be expected to be practically the same for the two cases.

The most valuable part of the paper undoubtedly is the observations on the drop size. The writer would appreciate seeing a curve showing how the size of the droplets varied along the nozzle, as well as a mathematical or, if not otherwise possible, an empirical relation between the important variables for calculation of the size of the droplets.

The discovery that there is a Wilson zone rather than a Wilson line should lead to a further analysis of the factors determining the initial point of condensation. There is need of a mathematical or graphical relationship which will enable us to calculate the point of condensation for any nozzle or blade. An important variable in this relationship apparently is the rate of expansion, or more clearly, the rate of change of velocity dV/dt, where V = steam velocity, and t = time.

J. I. Yellott.[7] The author in this interesting paper has contributed several valuable points to our knowledge of the manner in which flowing steam condenses. The difference in the location of the Wilson line for the rough and the smooth nozzle indicates that our conception of one line as a supersaturation limit must be altered. A Wilson region must be substituted, even at the expense of further complicating an already difficult problem. The writer is at present working on this phase of the subject, using a convergent-divergent nozzle in which the angle of divergence can be varied while the flow is being observed. The results of this work will be published in the near future.

It is believed by the writer that the author's location of the Wilson line for the smooth nozzle is too high. A great number of tests conducted with similar apparatus both at the University of Rochester and at Stevens Institute indicate that the Wilson line for a rapid expansion in a smooth nozzle should be located at about the moisture line of 4.5 per cent on the Mollier diagram.

The writer published in 1934 a Wilson line[5] which agrees with the author's, but later work indicated that an error had been made in the measurement of the initial temperatures. When this error is corrected, the writer's 1934 results agree with those obtained more recently, although both disagree with the author's. The writer is about to check his temperature measurements with a precision potentiometer, and it is hoped that the discrepancy will be cleared up.

Author's Closure

Mr. Coogan's comments concerning the method employed by

[4] Cambridge, Mass.

[5] "Supersaturated Steam," by J. I. Yellott, Jr., Trans. A.S.M.E., vol. 56, 1934, paper FSP-56-7, p. 411.

[6] Research Engineer, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa.

[7] Assistant Professor of Mechanical Engineering, Stevens Institute of Technology, Hoboken, N. J. Jun. A.S.M.E.

the author in the determination of the time required for condensation to occur are not pertinent in their entirety. He is justified in stating that condensation time depends upon the length and contour of the nozzle as well as roughness, but on examination it will be seen that all of these factors are considered in the equation used for the calculation of the condensation time, since the D term in this equation is a function of nozzle length and contour.

Furthermore, Equation [2] of the paper may be applied to any nozzle, regardless of its length, wherein the isentropic change in enthalpy is a linear function of nozzle length. Nozzle A, shown in Fig. 1 of the paper, possessed this feature in the region where the time element was being ascertained. Therefore, in this instance, the use of Equation [2] was proper. However, it should not be used indiscriminately for any type of nozzle.

The difference in condensation times, as found by Mr. Coogan when applying the two methods of calculation, was due to his misuse of Equation [2]. When employed correctly it will render an accuracy in excess of that obtainable by graphical integration; for the latter method involves an approximation of the limiting value of a sum of terms and not an exact evaluation as does Equation [2].

A determination of the lower limit of time required for condensation has not been attempted by the author.

Mr. Emmons' remarks pertaining to the moisture line of 11 per cent on the Mollier diagram as the limiting position of the Wilson line are interesting in that they substantiate the author's finding of a variable location of the Wilson line. This limiting position, discussed by Mr. Emmons, will be more nearly approached the greater the rate of change of velocity of the expanding steam in a nozzle.

In response to Mr. Egli's question, the back pressures were maintained at the values shown principally because the data required for the location of the Wilson line were not influenced by this amount of overexpansion. That is, the Wilson line would have the same location on the Mollier diagram as though the expansion had proceeded under design conditions.

This high back-pressure condition is not entirely out of keeping with modern turbine operation. For example, in a reaction turbine with a Curtis wheel on the high-pressure end, similar events as depicted in Figs. 2,3,4, and 5 of the paper will prevail, although to a lesser degree in those nozzle groups designed for a smaller load than the particular one existing on the turbine at any given moment.

In regard to the tests on nozzle B, the second set of steam conditions was referred to as different for the purpose of contrasting it with the set originally chosen.

The author has performed no experiments on the determination of droplet size variation. It is believed, however, that this may be accomplished by transmitting light of a known wave length through the flowing steam, and then applying the Rayleigh formula[8] deduced from Rayleigh's law of scattering. This experiment would require a glass bottom on the nozzle as well as a glass top.

The results of Professor Yellott's anticipated tests with the improved apparatus mentioned by him are awaited with interest by the author. Undoubtedly he will reveal interesting and useful information concerning the phenomena of supersaturation.

The Interpretation of Creep Tests for Machine Design[1]

Joseph Marin.[2] During the past year two theories have been developed for the design of member subjected to combined stresses accompanied by creep. One, a semiempirical theory, was formulated by Bailey,[3] while the other, based on the work of Saint Venant, has been developed by Soderberg.[1] A comparison of these two theories seems appropriate. In this discussion the comparison will be made between stress σ and creep rate C for simple tension on the basis of the power-function relation

$$C = A\sigma^n \qquad [1]$$

where A and n are experimental constants. It should be noted that the creep rate C is the constant minimum plastic strain or creep per unit time.

For the case of combined stress let σ_1, σ_2, and σ_3 be the principal stresses and C_1, C_2, and C_3 the corresponding creep rates. Then the values of the creep rate C_1 by the Soderberg and Bailey theories are, respectively,

$$(C_1)_s = \frac{A}{2^{(n+1)/2}} [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{\frac{n-1}{2}} \times [(\sigma_1 - \sigma_2) - (\sigma_3 - \sigma_1)] \qquad [2]$$

$$(C_1)_B = \frac{A}{2^{(m+1)}} [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{m} \times [(\sigma_1 - \sigma_2)^{n-2m} - (\sigma_3 - \sigma_1)^{n-2m}] \qquad [3]$$

Similar equations can be written for the creep rates C_2 and C_3.

Comparison of Theories With Experiments. A comparison of these two theories with experiments shows that both theories are in equally good agreement. This comparison was made by the writer for tests on lead tubes subjected to torsion made by Jamieson;[4] and tests on lead tubes subjected to internal pressure with axial loading, and steel tubes subjected to axial tension with torsion made by Bailey.[3] It is apparent, however, that there is need for further experimental work on this problem. Further tests may give the same conclusion since the difference between the values of the creep rates by the two theories is comparatively small. A comparison of the creep rates for the two-dimensional case for all ratios of principal stresses shows that there is little difference between the theories. However, it is of more significance to determine the differences resulting in designs using the two theories.

Applications. Steam piping is an example of a thin-walled cylinder subjected to internal pressure and axial loading. A comparison of the required thickness for this case as determined by the two theories shows that the maximum difference in thickness is about 12 per cent. This value considers all possible stress ratios. The case of a thin-walled cylinder subjected to internal pressure and pure bending gives a corresponding maximum difference in thickness of about 25 per cent. For cylinders subjected to high pressures, the thickness is no longer small compared to the diameter. An analysis by the writer of the thick-walled cylinder subjected to internal pressure by both theories shows a negligible difference for values $n = 6$, $m = 2$, as recommended by Bailey for medium-carbon steels.

A comparison of the previously mentioned designs by the two theories indicates that it makes little difference which theory is used. An analysis of the available test results shows that each theory is in equal agreement. However, there is an advantage in the use of the Soderberg theory in that the expressions for

[8] Wien-Harms "Handbuch der Experimental Physik," 1928, p. 368.

[1] Published as paper RP-58-15, by C. Richard Soderberg, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Assistant Professor of Engineering Materials, Rutgers University, New Brunswick, N. J. Mem. A.S.M.E.

[3] "Design Aspect of Creep," by R. W. Bailey, *Journal of Applied Mechanics*, vol. 3, Trans. A.S.M.E., vol. 58, March, 1936, p. A-1, abridged from "Design Aspect of Creep," by R. W. Bailey, read before a meeting at the Institution of Mechanical Engineers, London, November, 1935.

[4] "An Investigation Relating to the Plastic Flow of Lead," by J. Jamieson, Ph.D. Thesis, University of Michigan, Ann Arbor, Mich., 1933.

the creep rates are not as complicated as in the Bailey theory. In addition, the Bailey theory requires the determination of three constants A, m and n in place of two. For these reasons the writer recommends the use of the Soderberg theory. However, it should be mentioned that more experimental work is necessary before a definite conclusion can be reached.

Charles Davenport.[5] In this discussion the writer will use the nomenclature which is given in the paper. Equation [16] of the paper

$$d\sigma = -s_1 \frac{e^{\sigma/s_1} - 1}{1 + Te^{\sigma/s_1}} dT$$

which the author integrates numerically step by step to obtain the relaxation of a bolt may be integrated exactly as follows:

Substitute $Y = e^{\sigma/s_1}$ in the author's Equation [16], and rearrange, giving

$$\frac{dT}{dY} + \frac{1}{Y-1} T = \frac{-1}{Y(Y-1)} \qquad [4]$$

a linear differential equation, which is easily integrated

$$T = -\frac{(\sigma/s_1) + C}{e^{\sigma/s_1} - 1} \qquad [5]$$

with the condition that at $t = 0$ (or $T = 0$) the stress $\sigma = \sigma_6$ the constant becomes $C = -\sigma_0/s_1$, so that

$$T = \frac{\sigma_0/s_1 - \sigma/s_1}{e^{\sigma/s_1} - 1} \qquad [6]$$

In order to obtain the time taken for relaxation from 50,000 lb per sq in. to 20,000 lb per sq in., substitute $\sigma_0 = 50{,}000$ and $\sigma = 20{,}000$ in Equation [6] of this discussion, which gives $T = 0.313$. Then Fig. 3 of the paper gives the time as 2150 hr.

The step-by-step process followed by the author gives the shorter time of 380 hr since it extrapolates the higher rate of flow at the beginning of each Δt over the Δt. The large difference in the answers is due to the flatness of the relaxation curve in Fig. 4 of the paper.

The plastic flow necessary to allow this reduction in stress is $\epsilon = (50{,}000 - 20{,}000)/(30 \times 10^6) = 10^{-3}$. Fig. 1 of the paper shows that the time required to produce this flow under a constant stress of 20,000 lb per sq in. is 2150 hr; exactly the same as the time required to produce a flow of $\epsilon = 10^{-3}$ in relaxation from 50,000 lb per sq in. to 20,000 lb per sq in. That this coincidence is not accidental is shown by transforming Equation [6] of this discussion as follows

$$T = \frac{\frac{s_1}{E}\left(\frac{\sigma_0}{s_1} - \frac{\sigma_e}{s_1}\right)}{\frac{s_1}{E}\left(e^{\frac{\sigma_e}{s_1}} - 1\right)} = \frac{\frac{\sigma_0 - \sigma_e}{E}}{S_e} \qquad [7]$$

or

$$S_e T = \frac{\sigma_0 - \sigma_e}{E} = \epsilon \qquad [8]$$

where the subscript e denotes the end or final condition.

Equation [8] of this discussion states that the time required in relaxation to produce the flow ϵ, which will allow the stress to relax from a high value σ_0 to the final condition σ_e, is exactly the same as the time required to produce this same ϵ in creep, under the constant low stress σ_e. Physically this cannot be so, because the high stress in the initial period of relaxation produces a more rapid flow than the constant, low stress σ_e in creep.

The error which seems to have crept into the argument lies in the manner in which the formula $\epsilon = ST$ has been applied to relaxation. This function has been obtained as a good empirical fit of creep data and as such contains no assumptions. A family of creep curves may be depicted as a curved surface in three-dimensional space on the coordinate stress σ, plastic flow ϵ, and time t. The creep curves are intersections of this curved surface with planes σ = constant. The author of the paper makes the tacit assumption that the relaxation curves lie on this same curved surface, which is not permissible, as the writer will show. The assumption creeps in Equation [13] of the paper, which equation is an ordinary differentiation of the function $\epsilon = ST$, thus staying on the surface. The term SdT represents motion on this surface along a constant-stress curve for a short time and is, of course, physically possible. The second term of Equation [13], $T\frac{\partial S}{\partial \sigma}d\sigma$, represents plastic flow as the stress is changed and time is held constant. We know that no plastic flow can take place in zero time; therefore, this term has no physical significance.

It can now easily be seen why the same time was obtained for both creep and relaxation. If the relaxation curves lie on the same surface as the creep curves, then the value of the time for a given final value of σ and ϵ depends only on the end point and is independent of the path of integration. The latter parts of the paper, particularly Equation [30], are based on the same assumption and are therefore subject to the same criticism. Once it is clear that the σ, ϵ, t surface of the creep curves must be different from the σ, ϵ, t surface of the relaxation curves, the determination of the relaxation surface from creep data alone becomes a logical impossibility without a detailed knowledge of the whole phenomenon involved. In the absence of tests any theory which leads to plausible results is as good as any other. The writer proposes to discuss two such theories.

First, let us assume that we may simply drop the second term of the author's Equation [13], which we saw had to be zero physically so that

$$d\epsilon = SdT \qquad [9]$$

Now substitute $S = (s_1/E)(e^{\sigma/s_1} - 1)$ and $d\epsilon = -d\sigma/E$ and integrate from the limit: $t = T = 0$, $\sigma = \sigma_0$, with the result

$$T = \log_e \frac{1 - e^{-\frac{\sigma_0}{s_1}}}{1 - e^{-\frac{\sigma}{s_1}}} \qquad [10]$$

Relaxation from $\sigma_0 = 50{,}000$ to $\sigma = 20{,}000$ lb per sq in. gives $T = 0.076$ and from Fig. 3 of the paper we obtain the time for the relaxation as 60 hr.

Writing Equation [9] of this discussion introduces an assumption of how we drop from one constant-stress curve to another as the stress relaxes. This is illustrated in Fig. 1 of this discussion by the jagged line 1,2,5,6, which of course becomes smooth as the intervals are taken small enough. Equation [9] of this discussion sums up all the plastic flows that take place along the portions of creep curves as 1-2 in Fig. 1 of this discussion, while the vertical drops from one constant-stress curve to the next (2-5) at constant time are not added because they give no plastic flow. These vertical portions correspond to the second term of Equation [13] of the paper.

The second theory of arriving at relaxation curves from creep curves, which has been developed by Nádai[6] and other writers, is

[5] Harvard University, Cambridge, Mass.

[6] "The Creep of Metals—II," by A. Nádai and E. A. Davis, *Journal of Applied Mechanics*, vol. 3, no. 1, Trans. A.S.M.E., vol. 58, 1936, p. A-1.

shown as 1,2,3,4 on Fig. 1 of this discussion. It is based on the assumption that the stress is a function of the flow ϵ and the flow rate u, that is, $\sigma = f(\epsilon, u)$. This determines the manner of moving, from one constant-stress curve to another. When the plastic flow 1-2 shown in Fig. 1 of this discussion has taken place the

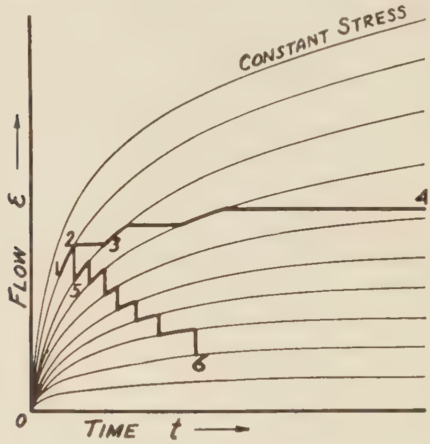

FIG. 1 TWO METHODS OF OBTAINING RELAXATION FROM CREEP
[Curve 1, 2, 5, 6, is for $d\epsilon = SdT$. Curve 1, 2, 3, 4 is for $\sigma = f(\epsilon, u)$.]

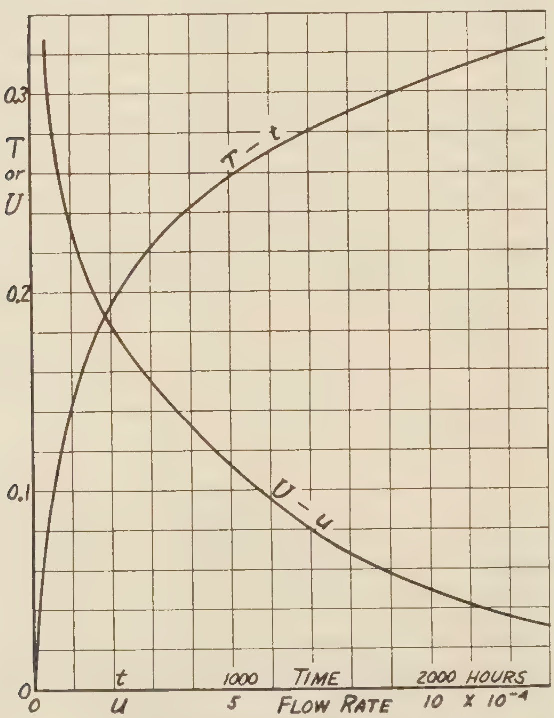

FIG. 2 FLOW-RATE FUNCTION U

stress is reduced and now this value of σ and ϵ determines the point 3. This includes strain-hardening effects but neglects the influence of time upon annealing or age hardening. Since this assumption $\sigma = f(\epsilon, u)$ seems the best and most reasonable existing, it will be interesting to find the relaxation time, combining it with the extremely good fit of the creep curves $\epsilon = ST$ as given by the author.

The slopes (flow rate u) of Fig. 3 of the paper are found and plotted against T which shall now be called U, since it represents flow as a function of flow rate, whereas T represented flow as a function of time. Just as a family of creep curves are obtained from Fig. 3 by multiplying the ordinates by S, a family of U-u curves can be obtained by multiplying both ordinates and abscissas of Fig. 2 of this discussion by S. So we can now write

$$\epsilon = SU \qquad [11]$$

This is simply a restricted way of writing $\sigma = f(\epsilon,u)$ which is a surface in space using stress, flow, and flow rate as coordinates. The assumption made in this theory is that the σ, ϵ, u surface is the same for creep as for relaxation. (Then the σ, ϵ, t surfaces for the two cases are different.) We now proceed to differentiate the relation, giving

$$d\epsilon = SdU + U\frac{\partial S}{\partial \sigma}d\sigma \qquad [12]$$

Substitution of $S = (s_1/E)(e^{\sigma/s_1} - 1)$ and $d\epsilon = -d\sigma/E$ yields

$$d\sigma = -s_1\frac{e^{\sigma/s_1} - 1}{1 + Ue^{\sigma/s_1}}dU \qquad [13]$$

which is similar to Equation [16] of the paper. Integration gives a result similar to Equation [6] of this discussion, or

$$U = \frac{\sigma_0/s_1 - \sigma/s_1}{e^{\sigma/s_1} - 1} \qquad [14]$$

Solve Equation [14] of this discussion for several values, pick the corresponding values of u from Fig. 2 of this discussion and multiply by S, giving values of the actual flow rate. Plot the reciprocal of this flow rate $dt/d\epsilon$ against the corresponding flow $d\epsilon = (\sigma_0 - \sigma)/E$. The area under this curve is the relaxation time. Relaxation from 50,000 to 20,000 lb per sq in. is found to require 450 hr.

Summarizing the results we find that the Soderberg numerical integration gives 380 hr, the Soderberg exact integration gives 2150 hr, the constant time drop between creep curves $d\epsilon = SdT$ gives 60 hr, and the constant flow (Nádai) $\sigma = f(\epsilon, u)$ gives 450 hr. The last method seems to be the most logical one devised to date. Soderberg's simple ST formula describing the results of creep tests considerably simplifies the application of Nádai's method to any specific case. However, the whole question of the relation between creep and relaxation data cannot be solved by logical deduction, and can be cleared up only by tests such as are now being conducted in several laboratories.

A. NÁDAI.[7] The analysis of the creep problem under combined stress presented by the author, and the various applications of it to a number of practical cases, deserve in many respects the attention of steam-turbine designers and those who wish to understand better the mechanical laws which must be valid for creep. Scarcely anything can be added to the concise manner in which the fundamental equations of flow have been introduced in the paper. Equations [1] to [8] inclusive, of the paper, contain what is essential for an analysis of problems of this kind under combined stress.

The author states that although there is a large amount of experimental information available on creep curves, the precise shape of these curves is not yet known, and therefore the mathematical formulas for their expression are not available. Under these circumstances, the author prefers to assume a special law in Equation [9] of the paper, expressing the shape of these curves. Among the various cases there is one which is of considerable interest, namely, when the creep curves $\epsilon'' = f(t)$, where ϵ'' is

[7] Consulting Engineer, research laboratories, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Mem. A.S.M.E.

the plastic strain and t is the time, are obtained under a constant stress σ, they tend to straighten out with increasing time t, that is, when the creep curves have straight asymptotes. According to Equation [9] of the paper, these straight asymptotes would have to pass through a common point in the plane of the coordinates ϵ'' and t. This would follow from the product form of Equation [9] of the paper

$$\epsilon'' = f_1(t)f_2(\sigma) \qquad [15]$$

It occurs to the writer that this, for example, is the case for the flow of a pure viscous substance, where

$$\sigma = \phi(d\epsilon''/dt) = \phi u'' \qquad [16]$$

where ϕ is a constant coefficient of viscosity, and $u'' = d\epsilon''/dt$ is the rate of strain. Another example is creep occurring according to the equation

$$\sigma = \sigma_0[1 + ln(u''/u_0)] \qquad [17]$$

that is, if a logarithmic speed law is valid, where σ_0 and u_0 are material constants. The viscosity coefficient in this latter case is

$$\phi = d\sigma/du'' = \sigma_0/u'' \qquad [18]$$

which is inversely proportional to the strain rate. Thus, if the viscosity coefficient is a pure constant, as in Equation [16] of this discussion, or if it is a pure function of the strain rate u'' or of the stress σ as in Equation [18] of this discussion, no changes of the material properties (of the material constants on which creep must depend) with time t are required to produce the effects expressed by Equation [9] of the paper. Since some experimental evidence favors ordinary creep-time curves $\epsilon'' = f(t)$ approaching a family of inclined straight lines, the more general case in which these straight lines would be represented by the equation

$$\epsilon'' = \psi_1(\sigma) + \psi_2(\sigma)\, t \qquad [19]$$

that is, a family of straight lines, having in general an enveloping curve, could not be covered by the form Equation [15] of this discussion.

These remarks involuntarily carry the discussion to the following questions which have figured in many discussions on creep and about which no agreement has apparently been reached by engineers: Does a steady state of creep exist and shall the straight portions of the creep-time curves be considered as the important portions for extrapolation?

In connection with these questions it is of interest perhaps to write the expression of the increment of the yield stress σ. In problems of this kind the yield stress ordinarily is assumed as a function of the plastic strains ϵ'' and the plastic-strain rates $u'' = d\epsilon''/dt$. As suggested by the author, the influence of the time t upon the material properties may also be included. The fact that at elevated temperatures the yield stress (or the hardness H which is associated with it) can change with time t when the metal is not strained at all (when ϵ'' = a constant and $u'' = 0$) is expressed by the differential relation

$$d\sigma = \frac{\partial\sigma}{\partial\epsilon''}d\epsilon'' + \frac{\partial\sigma}{\partial u''}du'' + \frac{\partial\sigma}{\partial t}dt \qquad [19]$$

Writing for

$$(\partial\sigma/\partial\epsilon'') = \psi,\ (\partial\sigma/\partial u'') = \phi,\ (\partial\sigma/\partial t) = X \qquad [20]$$

then Equation [19] of this discussion can be written as

$$d\sigma = \psi d\epsilon'' + \phi du'' + Xdt \qquad [21]$$

where ψ is the modulus of strain hardening, ϕ is the coefficient of viscosity, and X is the modulus of softening or hardening.

The metal hardens if $X > 0$, but if $X < 0$ it softens at the time t. After dividing Equation [21] of this discussion by dt

$$\frac{d\sigma}{dt} = \psi\frac{d\epsilon''}{dt} + \phi\frac{du''}{dt} + X \qquad [22]$$

is obtained in which the differential quotient on the left side expresses the rate of increase of stress, while u'' is the plastic rate of strain. For example, for a constant-stress test σ = a constant, that is, for the ordinary time-creep test, $d\sigma/dt = 0$ and

$$\psi u'' + \phi(du''/dt) + X = 0 \qquad [23]$$

is obtained. One of the many uses of Equation [23] of this discussion is for defining the conditions under which a steady state of stress is possible in creep. Equation [22] of this discussion offers an additional method for an approach to the problem of correlating the various special types of tests with each other, as for example, in discussing the constant-stress test, the relaxation test, or the test for constant-strain rate under given conditions.

The writer noted from the curves representing the distribution of the radial and tangential stresses, computed by the author for the case of a revolving disk containing a central hole, that the stress distribution after a certain time has elapsed does not change much more with time. In other words, after a certain time has elapsed and the stresses have redistributed themselves, they do not change further and seem to become steady.

The writer believes that the condition under which a state of stress is to be expected with a sufficient degree of accuracy is when the creep curves for a constant stress become straight lines.

Author's Closure

Mr. Marin's comments on the relative differences of the two theories are interesting and of practical value. The author certainly subscribes to the urgent need for more reliable tests of creep under combined stresses. However, the laws of plastic flow, which have been established for ductile materials at room temperature, and which really form the basis for the author's proposed theory, have a sound background of experiments. The application to materials at high temperatures is made difficult chiefly because of the difficulty of interpreting the results of the long-time creep tests themselves. No theory of combined stresses can be subjected to a really conclusive proof until it is possible to obtain curve fits superior to those now obtained by the power function used in this comparison by Mr. Marin.

The author is compelled to subscribe to the essential correctness of Mr. Davenport's criticism of the use which has been made of the empirical Equation [9] of this discussion. In the latter part of the discussion, however, it seems that Mr. Davenport has repeated some of the mistakes which the author so regretfully acknowledges.

Equation [9], $\epsilon = ST$, has been postulated as an empirical relation, which is valid only for paths of constant stress. The fundamental error committed by the author lies in disregarding this restriction in the differentiation of this equation, that is, Equation [15] of this discussion.

However, the author has not found it possible to agree with Mr. Davenport that there is more than one method of extrapolation from the constant-stress creep results to the relaxation case. The discussion of this point is simplified if T is treated as the real time variable, so that the creep curves are straight lines.

The validity of Equation [9] of the author's paper is en-

tirely a matter of experiment. Once accepted, however, it leads to the equation

$$\left(\frac{d\epsilon}{dT}\right)_{\sigma_{\text{const}}} = u = S \qquad [9a]$$

This relation expresses the fact, implicit in Equation [9], that the creep rate does not depend upon T or ϵ. An attempt to write the relation given by Equation [11], postulated by Mr. Davenport, thus leads to Equation [9a] of this discussion, and the two solutions must be identical. The differences obtained by Mr. Davenport must be ascribed to inaccuracies in the mechanical integration of Equation [14] of this discussion.

Now, if in the time dT the specimen is given a plastic elongation $d\epsilon$, the corresponding stress is uniquely determined by Equation [9a], or its solution in σ. Conversely, if a constant stress σ is applied over an interval dT, the resulting deformation is Sdt, regardless of the values of ϵ and T.

On this basis, it seems that the solution presented by Equation [10] of this discussion is the one which logically follows from the empirical Equation [9] of the paper. Experimental data are needed to prove or disprove this relation, but once it is accepted there does not seem to be more than one interpretation. Direct relaxation tests are difficult to conduct accurately enough to decide the point under discussion.

Mr. Davenport has pointed out a mistake which invalidates a certain part of the paper. It applies equally well to the differentiation in Equation [30] of the paper, and the step-by-step solution of the ring problem. If the Equation [9a] be used as a basis, the change brought about by Mr. Davenport's criticism is a considerable simplification of the entire process. Since the ultimate steady state is not affected, the modification of the result given by Fig. 7 of the paper appears to be less far-reaching than one would expect at first. A revision of this solution will be published later in connection with a planned discussion on the steady state of plastic flow.

Dr. Nádai refers to an important aspect of the plastic flow in two or three dimensions, namely, the premises of the steady state. The steady state is characterized by constant stress, even though the plastic flow may continue. The author does not consider the steady state related to the question of whether or not the creep curves for constant stress are straight lines. The basis for this assertion is that the essential aspects of the phenomenon portrayed by Fig. 7 of the paper are not changed at all, even if the creep curves at constant stress were straight lines throughout the entire flow. The only difference would be a difference in labeling of the individual curves of stress distribution.

The author regrets that the paper is not sufficiently explicit on this important subject. Equations [50] of the Appendix of the paper give the change in stress ($\Delta\sigma_{1r}$ and $\Delta\sigma_{2r}$) at any point r of the ring, as a result of a certain distribution of plastic flow (expressed through $\Delta\sigma_{1\xi}{}^*$) over the entire ring. The steady state is defined by $\Delta\sigma_{1r} = \Delta\sigma_{2r} = 0$. With this restriction Equations [50] become a pair of integral equations for determining the corresponding distribution ($\Delta\sigma_{1\xi}{}^*$) of plastic flow. Equation [59] of the paper is in fact the solution to these integral equations, and forms the basis for the plotting of the steady state.

So far, the steady state is not even related to a creep law; it is merely a geometrical expression for the configuration of flow which will produce no change in stress. To translate this result into an actual stress distribution it is necessary to resort to a creep law and an equation of equilibrium.

This part of the discussion of the steady state had to be deleted because of lack of space, but it will be referred to again in a future paper.

Supervising Instruments for the 165,000-Kw Turbine at the Richmond Station[1]

C. Richard Soderberg.[2] The original suggestion for remote recording instruments for large turbines appears to be due to G. L. Knight of the Brooklyn Edison Company and dates back several years. At that time outdoor constructions of large turbine aggregates were under serious consideration, and such turbine generator units would require a new approach to the problem of operation. While this scheme of powerhouse construction never matured into actual projects, it left a permanent mark upon conventional constructions in the form of a slightly modified set of supervisory-control instruments, of which the paper[1] gives the Westinghouse version.

The author's description of a development, to which he personally has made the major contribution, leaves very little to be required in the way of additional comments. It will be noted that, with the exception of the noise meters, the supervisory instruments of the Richmond unit are all based on the same fundamental measuring principle, with a common primary high-frequency circuit serving all of the elements. This principle, which is by no means original with this development, has proved to be eminently suited to the particular problems involved. The operating results have on the whole been quite satisfactory with the exception of those of the eccentricity meter. In the design of this particular instrument an attempt was made to obtain a device which would operate without mechanical contacts with the shaft. In addition, it was considered desirable to locate the instrument in the gland space, where the ordinary dial-gage readings are taken during slow rolling.

The location of electrical windings and a magnetic circuit just outside of the gland space naturally imposed severe requirements on their construction. While these difficulties undoubtedly could be overcome, the large displacements of the shaft in the oil film made it difficult to interpret the readings. On the basis of the experience at the Richmond Station, it has been concluded that an eccentricity meter which contacts the shaft is a more practical solution. Such eccentricity meters have since been constructed by the Westinghouse Research Laboratory at East Pittsburgh, Pa., and have proved to be of great value for field balancing applications. These have been of a portable type, however. A permanent eccentricity meter is not at the present time considered as a worth-while complication, since the routine data obtained by the remote recording vibrometers and expansion meters give the desired information.

The noise meters have given no particular trouble, but their continued use in the future is by no means assured, because of the uncertainty in interpreting the sounds obtained. The experienced turbine operator appears to prefer the simple listening stick.

With regard to the future applications of supervisory instruments of this kind, it may be stated that they can be justified, in the present form at least, only for giant installations in the class of the Richmond unit. A simpler and less expensive form of the vibration and expansion meters might possibly find applications on intermediate sizes of central-station turbines. The Richmond installation has been of great significance, however, in revealing expansion and vibration characteristics of large high-temperature turbines which would not have become known through the ordinary type of instruments.

[1] Published as paper FSP-58-9, by H. Steen-Johnsen, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Manager, turbine division, Westinghouse Electric & Manufacturing Company, South Philadelphia, Pa. Mem. A.S.M.E

(The following discussion was presented jointly with the discussion of the paper by J. L. Roberts and C. D. Greentree.[3])

T. C. Rathbone.[4] The expansion meter, as shown in Fig. 2 of Steen-Johnsen's paper, is mounted on one side of the thrust pedestal. The writer knows from experience that the pedestal does not always move out uniformly, but has a tendency to cock, that is, one side will get ahead of the other. In fact, on one particular unit the writer observed that this cocking was sufficient to cause binding in the gibs which resulted in a spasm of vibration. Until the trouble was corrected, it was necessary to free the pedestal by a sledge-hammer blow in order to stop the vibration. The point to be made is that if the expansion meter cannot be placed on or near the center line, two meters are required, one at each side, to show the true expansion movement.

Both shaft truth indicators described are of the magnetic-air-gap type. Some difficulty has been experienced in calibrating one of the systems. The writer would like to call attention to the fact that his company is making a practice of magnetizing turbine rotors for the purpose of carrying out magnaflux tests for fatigue cracks in blades, disk wheels, and rotor shafts, and up to now has not attempted to demagnetize the rotors. If the residual magnetism, which may not be uniform circumferentially, will have any effect on the eccentricity records, it will be necessary to demagnetize rotors equipped with such instruments.

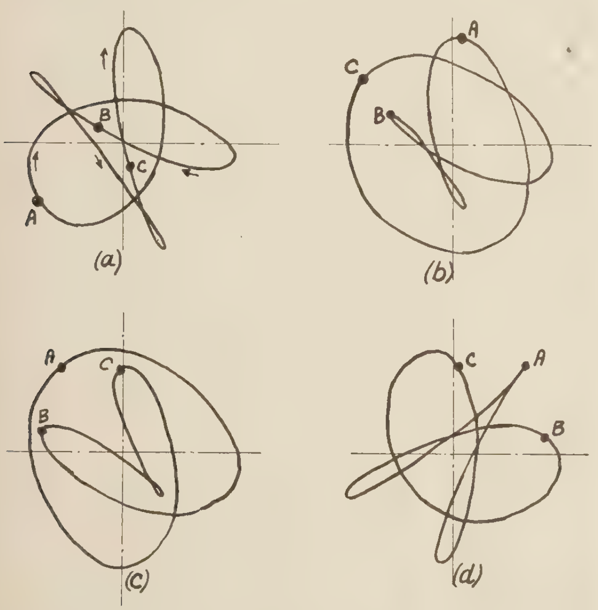

Fig. 1 Theoretical Vibration Figures Resulting From Combined 1200-Rpm and 1800-Rpm Influence of Four Simple Phase Relations 90 Deg Apart

(Lengths A to B, B to C, and A to C are each traversed in $^1/_{30}$ sec., the full curve is traversed in 0.1 sec.)

The eccentricity recorder described by Messrs. Roberts and Greentree is also used to record the shaft movement at the operating speed. The charts exhibited show that no relation exists between the eccentricity of the shaft and the bearing vibration. This is entirely out of line with the writer's experience. While in charge of vibration research at the South Philadelphia works of the Westinghouse Electric & Manufacturing Company, the writer conducted a rather exhaustive investigation of turbine vibration, and later published the results.[5] The principle that the vibration at the bearing responds to the rotor unbalance was confirmed by this investigation to such an extent that he was able to predict not only the amplitude, but the actual shape and phase relation of the vibration ellipse figure at each bearing for any combination of unbalance in the rotor. So far as bearing vibration is concerned, there is no difference between unbalance due to eccentric balance weights and that due to shaft eccentricity.

The erratic eccentricity readings may have been due to vibration transmitted from other machines, which affects the shaft as well as the bearings. Interacting influences between a 1200-rpm and an 1800-rpm machine produce a complicated vibration figure, and if the component in one direction only is being recorded, the

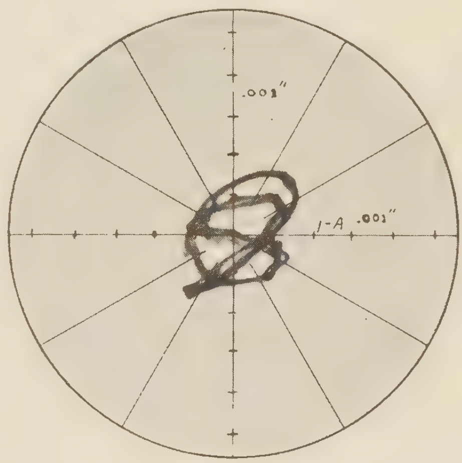

Fig. 2 Actual Record of the Theoretical Conditions Shown in Fig. 1

result would be very confusing. Fig. 1 of this discussion shows the theoretical vibration figures due to a combination of 1200-rpm and 1800-rpm influences, for four simple phase relations. Fig. 2 of this discussion shows an actual record of such a condition. By a suitable change in phase and amplitude, the observed figure could be produced synthetically. Observation of one linear component would be unintelligible.

The difficulty seems to be in the method of recording the eccentricity. The turbine and foundation are vibrating at the operating speed, and the only means for securing a steady reference point is the seismographic pendulum. If the eccentricity recorder be mounted on the bearing or casing which is vibrating, then it can no longer be a reliable reference point. If for example, the bearing vibration and the shaft movement were equal, but 180 deg apart in phase, the apparent eccentricity will be double the true amount. If in phase, no eccentricity will be apparent.

The eccentricity meter is a valuable and reliable instrument for the low rolling speeds. For some years the Westinghouse Electric & Manufacturing Company has supplied ordinary dial indicators and brackets to be used for checking the shaft eccentricity before bringing the turbine to speed. Charts were also supplied so that the operator could predict some time in advance how much rolling time would be required before the unit could be put on line. Checking the shaft for trueness has been a standard operating

[3] "Turbine Supervisory Instruments and Records," by J. L. Roberts and C. D. Greentree, Trans. A.S.M.E., vol. 58, November, 1936, paper FSP-58-7, pp. 607–614.

[4] Chief Engineer, turbine division, Fidelity & Casualty Company of New York, New York, N.Y.

[5] "Turbine Vibration and Balancing," by T. C. Rathbone, Trans. A.S.M.E., vol. 51, 1929, paper APM-51-23, pp. 267–284.

procedure for years,[6] so that nothing new is contributed by the elaborate recording devices described which suddenly make it possible to reduce the time necessary to get the unit on line, as the paper by Messrs. Roberts and Greentree would indicate. The contribution lies in the development of a means for recording the data at a remote point. Incidentally, with the unlimited available force supplied by the eccentric shaft, it would seem that the eccentricity record could be accomplished without the necessity for the elaborate and delicate apparatus described.

The spindle rolling gear was first introduced for the purpose of preventing spindle distortion. This accomplished two purposes: First, the time required to bring the spindle to speed was greatly reduced, and second, the danger of blade-tip and packing rubs during the first revolution was largely eliminated. But part of the danger of such rubs is due to the humping of the cylinder during shut down. It would be very desirable if some type of apparatus were developed which would indicate this cylinder distortion. Admittedly, the problem is difficult.

The writer has not had an opportunity to listen to the sound-transmitting devices, but believes that operators will have to be given some time to learn to differentiate the various noises, just as it requires a long experience to interpret the noise coming through the common listening rod. Attempts to amplify the noise by diaphragm or stethoscope attachments are usually confusing, after one is accustomed to the plain rod.

In this connection, it is suggested that further investigation may be profitable in the attempt to get warning of rubs by means of an electric circuit. On units equipped with the oil governor, there should be no metallic contact between the rotor and the stationary parts. The rotor is effectively insulated by the oil film. By means of a brush contacting the shaft at the No. 2 gland space, and a battery circuit with headphones, the writer was able to tell instantly when a micrometer mounted in the cylinder touched the rotor during operation. A blade or packing rub should have produced the same result. If all rotors were effectively isolated, this method for sounding an alarm would be far more positive than depending on sound.

Both vibration meters described are valuable achievements. With such apparatus it is possible to correlate changes in vibration with changes in operating conditions. The clues to many vibration difficulties have been uncovered by such studies.

The impetus for the development of the apparatus described was largely due to G. L. McKnight of the Brooklyn Edison Company. The writer recalls during preliminary discussions at the time, that a vibration alarm device was also proposed. Such an instrument mounted permanently on the turbine bearing could be set to sound an alarm when the vertical vibration reached some predetermined amount, say 8 mils. Few operators are around the units in a large plant. A saving in only 10 or 15 minutes in getting a unit in distress off the line might make a tremendous difference in the resulting damage. The writer still believes that such a costly machine as a steam turbine deserves being equipped with a vibration alarm.

John McLaren.[7] The authors of these papers[1,3] should be commended for a clear exposition of the possibilities of supervising instruments on turbine installations. The writer believes there is a large field for the use of instruments of this type throughout the country, not only in installations of large-capacity turbines but also in installations of smaller size where it is important, from a financial point of view, to have definite knowledge of the operating characteristics of the machines in order to forestall any condition which may result in severe rubbing or failure, and thus require expensive repairs or replacements.

The records shown by Messrs. Roberts and Greentree on the 160,000-kw and 20,000-kw turbines, and, more particularly, the variation between the eccentricity of the shaft and the vibration of the bearings are very interesting. The writer has supervised and analyzed a large number of vibration tests on both the bearings and the shafts of turbines, and has found that a study of readings from these tests generally gives a more accurate indication of the operating conditions of the machine than could be obtained by measuring the vibration amplitude from the bearings alone. Cases have been found where the vibration or eccentricity of the shaft was considerably greater than the vibration read from the pedestal, and it is believed that readings taken from the shaft will give an earlier indication of defective operation than will a vibrometer reading from the pedestal. This is evidenced from the records shown by the authors, wherein the shaft eccentricity went as high as 0.014 in., while the vibration amplitude of the bearing was of small magnitude.

In any possible development of these instruments for recording shaft eccentricity, the writer believes that it would be advisable to have these instruments located close to the bearing on the machine side of the bearing rather than on an extension to the shaft, because the readings can be too easily misinterpreted if they are taken on an extension to the shaft. The shaft itself, while running, will not always take up the same position in the bearing. The position of the shaft in the bearing will vary according to the load and the speed, with the result that when starting up on changing load conditions, unless continual compensation is resorted to, there is a possibility of the readings not being readings of the true eccentricity of the shaft.

The writer is somewhat disappointed that Mr. Steen-Johnsen did not show a record of the shaft eccentricity which had been taken with his company's instrument. It is unfortunate that these experiments are not being continued, inasmuch as considerable light would be thrown on the relation between the eccentricity of the shaft and the vibration of the bearings.

It is granted that a vibration record taken on the bearing alone is an important indication of the mechanical condition of the turbine, but an analysis of such a record in conjunction with the shaft vibration would be a greater help to the operating engineer and be of valuable assistance in an analysis of any trouble which may occur.

The interference detector or noise meter should also be a valuable adjunct to supervisory instruments in the operation of turbines. While an experienced operating engineer can determine, with a considerable degree of accuracy, faulty operation of a turbine by use of the listening rod, even the most experienced operator is handicapped by the limited and variable sensitivity of the human ear. There is a large number of turbine installations throughout the country which have been in operation for a number of years. Many of these units have undergone repairs from time to time, and if noise-meter records were available for such units, they would be a valuable indication of the condition of these turbines not only while starting up or while increasing or decreasing the load, but also in regular operation. The writer does not believe that the average operator would have any difficulty in analyzing the noise records once they were thoroughly understood. It is realized the record looks rather complicated, but if it were studied along with the other records that have been discussed, valuable information could be obtained.

The writer believes that in considering the further use of the noise meter and bearing vibrometer, it would be worth-while to install an oscilloscope which could be connected with these instruments and switched in when records indicated there was something wrong. The oscilloscope is being used more and more and

[6] "Turbine Shaft Distortion," by T. C. Rathbone, *The Electric Journal*, Feb., 1931, pp. 91–95.

[7] Supervising Engineer, The Travelers Insurance Company, Hartford, Conn.

it is only a matter of time until this instrument also will be used very effectively in conjunction with supervisory instruments. After the fundamentals are learned, the average operating engineer should have no difficulty in interpreting the figures shown on the screen of this instrument. It would not add greatly to the expense of the supervisory instruments and would be a valuable addition.

In the writer's opinion, there is a wide field of usefulness for properly designed and simplified supervisory instruments on turbine installations of a size much smaller than has been contemplated in these papers.[1,3]

Author's Closure

Some interesting features of the turbine-vibration problem are given consideration by Messrs. T. C. Rathbone and John McLaren in their discussion of the author's paper.

Mr. Rathbone calls attention to the tendency of the thrust pedestal to cock, with unequal cylinder expansion, and the consequent need for two expansion detectors in order to record fully the expansion characteristics of the turbine.

The author does not believe this refinement to be necessary for everyday observations. Cocking of the pedestal as we know it is invariably accompanied by nonuniform expansion, which shows up as a jagged line on the expansion record. However, when it becomes necessary to conduct a detailed analysis of vibration trouble, the amount of cocking of the pedestal is of importance. In such an analysis it is therefore necessary to secure expansion recordings on both sides of the pedestal. In connection with a recent vibration investigation, two expansion detectors were used, and the recordings obtained were vital factors in locating the trouble.

Regarding the magnaflux testing, the residual magnetism in the shaft unquestionably will influence the readings from an eccentricity detector that utilizes the solid shaft as part of the magnetic circuit. However, if a laminated ring is mounted on the shaft and the magnetic circuit completed through this ring, rather than through the solid shaft, there should be no trouble from residual magnetism in the soft iron ring.

The inconsistency observed in the eccentricity and vibration recordings exhibited by Messrs. Roberts and Greentree is, as Mr. Rathbone points out, not in line with earlier observations. It is the author's opinion, that the inconsistency is caused by the location of the pickup on the shaft extension. The readings on each side of a bearing are seldom the same, and of opposite phase. As a rule, the shaft extension shows a smaller reading but larger readings also have been observed, and frequently the readings have been observed to be in phase; but in the majority of cases the phase relation is erratic.

As pointed out by Mr. McLaren, the most reliable readings are obtained close to the bearing and on the machine side of the bearing. It is true that the amplitude on the shaft is greater than on the bearing in the majority of cases. Particularly is this true on 3600-rpm high-pressure machines. But for the observation of day-to-day changes in a machine, and as a warning signal of unusual operating characteristics, the readings on the bearing are satisfactory, when obtained with an instrument of suitable sensitivity. On the other hand, for purposes of efficient balancing, or complete analysis of vibration trouble, it is vastly preferable to use shaft vibration.

It is the author's practice to obtain such shaft readings mechanically, with instruments that are flexibly supported. This avoids the reading of relative eccentricity between the pedestal and the shaft. An electrical eccentricity detector, as mentioned in the original paper, operating from a follower on the shaft, would be constructed on the same lines to give absolute readings. If the demand should warrant it, the development of such a device could unquestionably be carried through to success. However, the interest in instruments of this type in connection with active turbine installations does not indicate such a demand.

Mr. McLaren expresses interest in the relation between shaft and bearing vibration. At present, time does not permit entering into this question, but it will be considered in a future paper.

Turbine Supervisory Instruments and Records[1]

G. L. Knight.[2] Supervisory instruments received their first impetus some years ago when engineers of the Consolidated Edison System were seriously considering the installation of large outdoor turbogenerator units in a proposed new generating station.

At the time various alternative power-station designs were considered, but in general the suggested design involved installing the boilers and turbogenerators in the open air with a service building between them. Such equipment as condenser auxiliaries, feedwater pumps, and fans were to be housed in basement or ground-level structures. Control of the turbogenerators was to be concentrated in the basement of a service building adjacent to the turbine-room basement.

Such a design called for four different types of remote-control indicating and recording instruments which would give the operator reliable and constant indication of the condition of his turbine, both in starting and shutting down and in regular operation. It also involved the operation of the throttle valves by some form of Selsyn motors with position indicators and suitable oil- and steam-temperature indicators, all of which would have been comparatively simple since applications of these devices and instruments had already been worked out.

For the remote instruments it was necessary, as Messrs. Roberts and Greentree have pointed out in their paper, to have first, an instrument for indicating the straightness of the turbine shaft measured by eccentricity at the end of the shaft; second, an indication of vibration at several critical points on the turbine; third, an indication of unusual noise, that is, other than the steam flow, which would give instant notice to the operator at the start of any rub; and fourth, an indication of the expansion of the turbine shell at starting and loading time, and also at the time of putting the unit back into service after overhaul.

An order was given to the General Electric Company for a set of these four instruments to be installed on the No. 8, 160,000-kw General Electric turbogenerator at the Hudson Avenue Station with the understanding that the instruments were to be paid for when accepted as satisfactory. While it has taken a much longer time to develop the instuments in both factory and field than was anticipated by either the General Electric Company or the Brooklyn Edison Company, they have now been so developed and accepted, and are being placed in everyday operating service.

The outdoor type of station has been given up, because further analysis of station design showed only a very small difference in cost per kilowatt of capacity between outdoor and simplified indoor types. However, the instruments as now developed are considered a distinct forward step for the completely housed station because they give the turbine operating engineer a definitely superior and more reliable guide for indicating the condition of the turbine at all times. Thus these instruments have promoted safer operation, and in turn, improved reliability of service.

[1] Published as paper FSP-58-7, by J. L. Roberts and C. D. Greentree, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Vice President, Booklyn Edison Company, New York, N. Y.

The Brooklyn Edison Company will be glad to give an account of operating experience with these instruments and suggests that time be devoted at some future meeting to accounts of the operating experience both with these instruments and those developed by the Westinghouse Company and installed on the 165,000-kw turbogenerator at the Richmond Station of the Philadelphia Electric Company.[3]

To the authors of the paper belongs the credit for overcoming many obstacles in the development of these new and better tools and finally bringing them to the state of perfection which justified their acceptance.

C. Richard Soderberg.[4] This paper, with the companion paper by Mr. Steen-Johnsen[3] on the same subject, presents an interesting example of two independent solutions to a common problem. The methods employed differ only in minor details, and the operating experiences appear to have been quite similar.

The data given from actual turbine operation are of considerable interest in that they reveal the typical behavior of large turbines not previously recorded in this manner. The authors note that, once the eccentricity of the turbine shaft has been reduced to a certain minimum, the remainder of the starting period can be much abbreviated. This is an important subject, on which it is difficult to obtain reliable information, since most operators quite appropriately are not inclined to experiment with short starting periods just for the sake of scientific information.

The importance of the turning gear in the operation of large turbines cannot be exaggerated in this connection. The turning gear was introduced not so very long ago as a convenient but by no means indispensable aid, particularly for units subjected to frequent starts. The experience obtained on the Richmond unit,[3] which due to peculiar load conditions is started and stopped daily during certain periods of the year, is even more remarkable than that recorded by the authors for the units at the Hudson Avenue Station. There is no exaggeration in stating that such a mode of operation would be entirely impractical if the turbine were not equipped with a turning gear.

The use of supervisory instruments of the type described is certainly justified on very large units, particularly during the investigation of obscure vibration phenomena. The same results could conceivably be obtained by portable instruments of simpler construction, brought to the station during the investigation. It would be interesting to have the author's opinion with regard to the justification for this degree of complication as a standard accessory for central-station turbines in the moderate sizes.

Authors' Closure

Several pertinent points in connection with turbine supervisory instruments have been high-lighted by the foregoing discussions, and certain features of instrument design, which were not adequately covered in the original paper, have been questioned.

G. L. Knight's proffered subsequent account of operating experience with these instruments should furnish valuable information and the authors hope that plans can be made to include such a report at some future meeting of the A.S.M.E. The General Electric Company is now equipping several turbines with these devices, and much additional information on the behavior and operation of several types and sizes of turbines will shortly be available.

In reply to C. R. Soderberg's concluding question as to the value of these instruments on moderate-size turbines, the authors believe that their protective functions in addition to their aids to operation make them economically justifiable on turbines of 10,000 kw, or even less, when it is desired to maintain turbine efficiency and restrict outages. This is especially true of the eccentricity recorder which, it will be remembered, supplied correct information quite at variance with that which might have been deduced from the vibration records only, or from vibration readings by an operator. True, instruments simplified by the substitution of indicating for recording meters, could be brought to the station for the investigation of obscure vibration phenomena, but these would hardly substitute for the continuous, day-by-day record on which any deviation from previous patterns will be immediately evident to the operator. When conditions make it necessary or desirable to shorten the starting period, an operator of even a small turbine, equipped with these instruments, can adapt his practice to the recorded conditions, rather than having to depend on experimentation or slow handmade measurements which he may not have time to make.

T. C. Rathbone's discussion presents some interesting points which can best be clarified by considering certain phases of the design of the instruments which were omitted from the paper.

While location of the expansion-detector unit on one side of the thrust pedestal might give erroneous information if there were binding in the gibs, the General Electric expansion detector can be (and is, on the 20,000-kw mercury turbine) mounted on the turbine center line.

It will not be necessary to demagnetize rotors equipped with supervisory instruments, because experiments have shown that random magnetic poles throughout the shaft have no effect on the eccentricity measurements made between the detector coils and a special ring located at the front end of the shaft.

Interpretation of the Brooklyn Edison eccentricity records on the basis of interference from structure-borne vibrations from adjacent synchronous or asynchronous machines clashes with the three following points:

1 Except during starting, there were no asynchronous machines adjacent to the Brooklyn Edison turbine on which the records were taken.

2 The structure-borne vibration from an adjacent machine, as shown at the right-hand end of the vibration-amplitude record Fig. 3 of the paper, is much smaller than the eccentricity values being measured. Furthermore, adjacent-machine load changes produced no discernible change in the measured machine records.

3 By means of a specially adapted seismographic type of indicator in actual contact with a smooth rotating surface concentric with the shaft center line, the electrically detected eccentricity-recorder readings were checked and verified point by point during a starting sequence and the subsequent loading period.

Furthermore, it must be remembered that the object of the eccentricity recorder is to obtain a true picture of the bending of the shaft relative to its own shell, and not relative to a fixed point in space. The latter procedure would encompass low-frequency foundation movements of which both the shaft and its shell may partake, thus needlessly and erroneously complicating the record. In the seismic instrument checking tests previously mentioned in this closure, the movement of the shell relative to a fixed point was so slight in comparison with the eccentricity being measured that the results were substantially identical on either basis.

True, vibration of the eccentricity-detector coils relative to the shaft-eccentricity ring will produce erroneous results, and for this reason great care is taken to mount these coils in such a manner that what small vibration they do have will be considerably less than even the small eccentricity values which can be recorded.

Mr. Rathbone's point that following rods with dial indicators have been used for some time to indicate shaft eccentricity at low rolling speeds is well taken. Yet their very limitation to the low

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[3] "Supervising Instruments for the 165,000-Kw Turbine at the Richmond Station," by H. Steen-Johnsen, Trans. A.S.M.E., vol. 58, November, 1936, paper FSP-58-9, pp. 621–626.

[4] Manager, Turbine Division, Westinghouse Electric & Manufacturing Company South Philadelphia, Pa. Mem. A.S.M.E.

rolling-speed range added impetus to the need for a continuous-recording device good at all speeds. As the published records so clearly show, large eccentricities may be produced at synchronous speed during the loading cycle, even though the eccentricity while coming up to speed was not excessive. This operating-range extension, together with the continuous-recording features, gives the operator a tool quite different from that which was formerly available.

Changing the form of listening device to which the operator is accustomed is admittedly risky, but the value of the interference detector lies in its tremendous amplification. Under loaded conditions, the turbine and steam noises are so complex and loud that only an experienced operator can distinguish those particular groupings of sounds beneath the general noise level which are associated with trouble. But at the time of greatest danger from rubs, when the turbine is starting up after a shutdown, the turbine noise is unencumbered by steam noise, and the available amplification makes slight rubs (which are indistinguishable with the conventional listening rod) not only audible but clear enough to be heard above the air-borne noise from adjoining machines. In addition, the inception of a rub is instantaneously announced to all the operators adjacent to the turbine, rather than having to depend on a single operator who may not be present when the rub first develops.

The vibration device described in the paper is also capable of tripping an alarm signal, as mentioned by Mr. Rathbone. One way which we have provided for tripping such a signal, when vibration amplitude exceeds a certain predetermined amount, is by the application of standard contact-making points to the recorder mechanism of the vibration-amplitude recorder described.

Replying to McLaren's discussion the authors would emphasize the fact that the actual shifting of the shaft during speeding up and loading sequences, to which he refers, is the condition which caused the most trouble in the development of the eccentricity recorder. The circuit of the eccentricity detector described in the paper is so constructed that the once-per-revolution, cyclical, true shaft-curvature reading is independent of the average position of the shaft in the bearing. With a shaft eccentricity of 0.0005 in., the average gap between the shaft ring and the detector coils can vary between 0.005 and 0.05 in. without changing the recorder reading of $^1/_2$ mil a visible amount.

In regard to the value of a noise record, it should be pointed out that noise tests on a large loaded steam turbine have shown that the noise spectrum is practically continuous from 60 to 5000 cycles per sec. That is, substantially every frequency is present at about the same relative intensity. Thus, it is questionable if any clear picture could be obtained with an oscilloscope. This, of course, does not hold when starting the turbine, that is, at low speeds and unloaded conditions.

Superposed-Turbine Regulation Problem[1]

C. Richard Soderberg.[2] The problem of control of steam turbines embodies practically all aspects of the theory of vibrations. The phases of greatest practical importance relate to the question of stability of systems with many degrees of freedom. The phenomenon of self-excited vibrations, which has been the subject of much study in other fields in recent years, constitutes one of the inherently natural properties of the systems encountered in the control problem and represents merely one form of instability.

While the authors do not present anything new in the way of fundamental approach, it is of significance in introducing the subject to the Applied Mechanics Division of the Society, and the writer wishes to express the earnest hope that it will lead to further contributions on the subject.

The form of hydraulic control to which the paper is applied is a comparatively recent modification of an older system which is now in operation on a very large number of turbines covering a wide range of capacities. The principal improvement consists in the introduction of the hydraulic transformer, whereby relatively weak impulses are stepped up without the usual handicaps of troublesome time lags. The inherent flexibility of this transformer system constitutes a real advance in the art, particularly when it is applied to more complicated installations of automatic extraction turbines. The quantitative analysis of this device, which is presented in this paper, has enabled us to avoid many of the pitfalls so common in development work of this nature, and represents a very real proof (if any be needed) of the value of applied science in mechanical engineering.

The particular problems encountered in the control of superposed turbine applications have more than ordinary significance at the present time. The superposed turbines now under construction embody several departures from more conventional turbine practice, all of which have served to accentuate the importance of the governing system. The inlet steam conditions are very high, most of the units being constructed for a pressure of 1200 lb per sq in. and a temperature of 900 F. The steam flows are enormous, reaching up to the order of 1,500,000 lb per hr. Most of the units are designed for operation at 3600 rpm, so that the inertia of the rotating system is inordinately small. As a result of these circumstances, the characteristics of these units introduce an entirely new order of regulation problems. This can be seen most clearly in the rate of acceleration at the moment of a sudden elimination of the load with the control valves wide open. Most of the turbines now in existence will accelerate at the rate of 100 to 200 rpm per sec under these circumstances, and most of our conceptions of the requirements of control systems have been formulated against this background of experience. The largest unit now under construction by the Westinghouse Electric & Manufacturing Company will accelerate at the rate of 765 rpm per sec. This means that an overspeed of 10 per cent will be reached in less than 1 sec, and if the overspeed protection fails to function a major disaster may be precipitated in a few seconds. The characteristics of the rotating system in units of this type are not unlike those of the projectiles in modern ordnance.

With these facts in mind, the authors have a perfect right to dare anyone to classify the mathematics, complicated as it happens to be, as of merely academic interest.

T. C. Rathbone.[3] The problem of regulation and stability in governing of superposed turbines in normal operation is ably treated in this paper. An additional control problem on high-speed high-capacity topper units meriting further consideration is that of confining the speed jumps on sudden loss of full load to within reasonable limits.

Mr. Soderberg in his discussion[2] calls attention to one of the inherent characteristics of the superposed turbine, that is, the combination of large steam flow and small rotational inertia, and advises that initial accelerations of more than 700 rpm per sec are to be anticipated on sudden loss of full load with some of

[1] Published as paper FSP-58-8, by A. F. Schwendner and A. A. Kuoma, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Manager, turbine engineering department, Westinghouse Electric & Manufacturing Company, South Philadelphia, Pa. Mem. A.S.M.E.

[3] Chief Engineer, turbine division, Fidelity & Casualty Company of New York, New York, N. Y.

the new designs. This compares with 100 to 150 rpm per sec on existing large condensing units.

These figures are somewhat startling. Some years ago the large-turbine manufactures were called on to provide governing equipment which would permit dumping full load suddenly without letting the speed rise reach the emergency-stop setting of 10 per cent. This was accomplished, with a small margin to spare, by improvements in the speed-responsive governor control relays. One case, however, apparently required an anticipator device to close the valves more quickly, because of a greater volume of trapped steam beyond the valves.

Under these newer and more drastic conditions, the question naturally arises as to whether or not the speed-responsive governor can function quickly enough to prevent excessive speed rise in the event of sudden loss of full load, and how much more dependence must be placed on the second line of defense, the emergency-stop governor and throttle valve.

Judging from the results of tests carried out by the Fidelity & Casualty Company by means of recording apparatus specially developed to determine both the accurate tripout speed and the time lag between the tripout and the closing of the throttle valve, there is some cause for concern. We are finding a number of cases of time lags reaching 2 sec. The average time lag appears to be about 0.7 sec. Two of the leading manufacturers limit the lag to 1 sec, and attempt to reduce it to 0.5 sec.

With an initial acceleration of 720 rpm per sec, 10 per cent overspeed is reached in 0.5 sec. On these new units with smaller Wr^2, the energy of the trapped steam will be even more effective in sending the speed higher after the valves are closed. Thus, allowing a time lag of 1 sec, it would appear that if the governing valves fail or are sluggish in function, the best that the average emergency-governor system can be expected to do is to let the overspeed reach values of between 30 and 35 per cent. In the past, none of the large-capacity units were tested in the shops to more than 20 per cent overspeed. It looks as though the same reliance cannot be placed on the overspeed trip as formerly, as a second line of defense.

On the rather elaborate load-dumping tests carried out some years ago by the Westinghouse Electric & Manufacturing Company in the development of control mechanism, the writer recalls that on the various fractional load dumps, the corresponding initial accelerations furnished a fairly accurate index of the maximum speed reached. In other words, the maximum speed to be reached later could be anticipated at the very instant the load was dumped. It was proposed at the time to take advantage of this fact by means of an inertia governor which would be responsive to acceleration rather than to speed. This governor would therefore anticipate the speed jump and function to close the valves before the regular governing system would be called on to act, if the acceleration exceeded some predetermined amount. Improvements in the standard governing system at the time made this additional apparatus unnecessary.

In view of the new conditions to be encountered, and from the standpoint of the insurance carriers, it seems pertinent to inquire whether some such anticipatory device might not be appropriate now, and also what is to be done to secure the same emergency-stop protection which has prevailed up to now.

Ronald B. Smith.[4] The problem of synchronizing instability in a turbogenerator set generally arises from one of two sources: An inherent instability, resulting from an improper relation between the time constants of the governing system, the rotating masses and the damping; or a forced instability brought on by friction in the operating mechanism or peculiarities in the valve characteristics. In the turbine field, until the present time at least, the instability problems have been mainly of the forced variety, but, with the advent of the lightweight high-capacity topping turbines the inherent instability problems which the authors discuss are coming to the fore.

There is a rather interesting rearrangement of the right side of the authors' Equation [9a] that seems justified in view of its clarity. Providing the slope differences between φ_{v1} and φ_v in the valve characteristic, as shown in Fig. 2 of the paper, are not too large, and the regulation is approximately linear, the torque constant T_M may be written

$$T_M = \frac{1}{\delta}\left(\frac{\dot{\omega}_0}{\omega_0}\right) \qquad [1]$$

where δ = the regulation of the governor, or the ratio of the change between no load and full-load speed, and the full-load speed $= (\omega_{NL} - \omega_0)/\omega_0$; ω_0 = full-load speed radians per sec; $\dot{\omega}_0$ = acceleration of the turbine generator on dumping full load, radians per sec per sec.

Under these conditions the authors' stability criterion becomes

$$\frac{1}{T_I + T_x} + \frac{1}{T_v\left(1 + \dfrac{T_x}{T_I}\right)^2} > \frac{1}{\delta}\frac{\dot{\omega}_0}{\omega_0} \qquad [2]$$

Thus, the closer the regulation δ, or the higher the accelerating rate, which is particularly the problem in superposed designs, the more difficult the inherent stability problem.

Consider a design in which the steam-chest volume is reduced to negligible proportions ($T_1 = 0$). This is always attempted in order to limit overspeeding on load dumps, and in industrial-type machines is generally realized. Under these conditions stability exists if

$$\frac{1}{T_x} > \frac{1}{\delta}\frac{\dot{\omega}_0}{\omega_0} \qquad [3]$$

A fast-operating transformer is the only necessity; in fact, if the transformer were removed ($T_x = 0$), and the mechanism worked properly, the system would always be stable. In so far as inherent stability is concerned a reasonable conclusion is that the operating mechanism is insignificant; for overspeeding quite the contrary, a fast-operating mechanism is important.

S. D. Mitereff.[5] The authors should be congratulated on an exhaustive quantitative analysis of time lags occurring within as well as outside an installation of steam-turbine speed governor. Time lags within the governor itself may be termed as residual inasmuch as evidently all possible efforts have been taken to reduce them to a minimum by a skillful application of practical expedients peculiar to the art.

The basic characteristic of the governor is $F = k_1P$, where F is the distance traversed by the throttle valve, P is the magnitude of speed change, and k_1 is a constant of sensitivity adjustment. Knowing the time-lag constants of installation, an optimum value of k_1 could be readly calculated from equations of the paper.

It would be interesting to know the actual magnitude of the time-lag constants of a typical installation. For instance, if the constant of time lag outside the governor is, say, three or more times as great as the combined time-lag constant of the governor itself, then the lag outside the governor will be a controlling factor in limiting the value of k_1 optimum, and no improvement in the governor will appreciably increase k_1 unless the terms $k_2(dP/dT)$ and $k_3(d^2P/dT^2)$ are first added to the basic characteristic of the governor.

[4] Turbine engineering department, Westinghouse Electric & Manufacturing Company, South Philadelphia, Pa. Jun. A.S.M.E.

[5] Y.M.C.A., Petersburg, Va.

It would be interesting also, if authors could give the comparison between the calculated k_1 optimum, and that obtained in practice. This point is important not only as a check on the equations, but also as an evidence that no time lag has been overlooked by the authors in setting up their equations.

The writer has in mind particularly the influence of inertia of steam in the supply line, which may be considerable with a lagging long line and modern superpressures, and also the influence of changes in the nozzle and bucket-wall temperatures as affecting steam conditions inside the turbine. He thinks that a slightly premature opening of nozzle admission valves has been adapted in order to minimize the "valleys" in relay-position–steam-flow plot occurring at transition points. A perfectly straight-line relationship can be obtained without loss of turbine economy by installation of a cam-type motion instead of present lever motion between relay operating piston and relay pilot-valve restoring spring. Such a cam will have to have irregular cavities corresponding to and compensating the above-mentioned valleys.

Ed S. Smith, Jr.[6] The authors have presented, in admirably compact form, an analysis of turbine regulation under particular conditions that broadly follows the method used by the writer in an analysis[7] of typical industrial regulators. This method involves the setting up of the differential equation for the regulated system, and obtaining the stability of the system from the equation's solution. This reference[7] is included as of value in tying together references in related fields.

Authors' Closure

The authors wish to thank Mr. Soderberg for his helpful comments on their paper.

Mr. Rathbone's discussion emphasizes the principal difference in initial accelerations of the superposed and large straight-condensing turbines and its effect on the shaft speed rise when load is suddenly lost.

Mr. Smith's stability criterion shows more clearly that the initial acceleration of the turbine shaft is part of the governor stability problem. Governors designed with the required characteristics will have sufficiently low time-lag constants to limit the maximum shaft speeds within normal values.

Turbine design specifications call for the governors to limit the maximum shaft-speed rise to that below the autostop speed when full load is dumped. Most of the present superposed turbines have a calculated maximum speed rise of 8 to 9 per cent when full load is dumped. Only on units with large steam flows will it be more difficult to maintain the speed within these limits. For these cases an electric anticipating device is used. The anticipator will close the governing valves immediately if load is dropped at a faster rate than that for which the anticipator has been set. This instrument has been tested and is in successful operation on a few installations. Even without the anticipator and assuming a somewhat sluggish governor, the governing valves will be partly or almost closed by the time the autostop speed is reached. The steam flow by that time is sufficiently throttled to keep the maximum shaft speed well under 20 per cent even under this adverse condition. The throttle valves furnished for superposed turbines have a closing time of less than one second between the tripping of the auto stop and the closing of the throttle valve. As an additional safeguard, the tripping of the autostop also closes the governor valves instantly, a practice the Westinghouse Company followed for a good many years on all large turbine governing valves. These precautions, with the volume of entrapped steam beyond the governing valves reduced to a possible minimum in comparison with the long steam lines between the governor valves and turbine on the old condensing units, will bring the superposed turbine to an equal if not better position than the old condensing units, as far as overspeed protection is concerned.

The authors are indebted to R. B. Smith for his suggested clarification and simplification of the torque constant T_m and the stability criterion. The authors do not agree with him about the possibility of reducing the steam-chest volume to such an extent that it can be completely neglected. His final stability criterion would mean that all governors so far built or conceived, as long as they are not burdened with a transformer, would be stable under all conditions as long as there is no steam-chest lag, a conclusion we are sure Mr. Smith does not intend to imply. The transformer was introduced to obtain an operating response to very small speed changes. This characteristic was described in a previous article.[8]

Mr. Mitereff is quite right in his assertion that in cases where the time lag outside the governor is three or more times greater than the combined time lag of the governor itself, a new characteristic has to be added to the governor to make the governing system stable. However, the effects of an acceleration element added to the velocity element in control problems represented by a turbogenerator is probably not as beneficial as Mr. Mitereff stresses. This can be shown by considering a case of a flyball governor with a turbine. With simplifying assumptions, the motion of such a system can be represented by two equations

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + Kx = K_1\omega \qquad [1]$$

$$J\frac{d\omega}{dt} = -K_2x \qquad [2]$$

Where m = equivalent mass of the governor, c = damping constant, k = governor-spring constant, x = displacement of governor sleeve from neutral position, K_1 = increase of force on the governor sleeve due to speed increase of ω rad/sec, $J = WR^2/g$ of turbogenerator, and K_2 = change of torque due to change of governor position.

If this system is to be stable, we must have

$$c > \frac{mK_1K_2}{KJ} \qquad [3]$$

The addition of acceleration element will make (again with some simplifying assumptions) Equation [1*a*]

$$m\frac{d^2x}{dt} + c\frac{dx}{dt} + Kx = K_1\omega + K_3\frac{d\omega}{dt} \qquad [1a]$$

The stability criterion for the system represented by Equations [1*a*] and [2] is

$$C > \frac{mK_1K_2}{KJ + K_2K_3} \qquad [3a]$$

Which inequality is satisfied easier, Equation [3] or [3*a*], depends largely on the practical consideration of the problem.

So far, with the present design as used on existing units, we did not find it necessary to use the characteristic mentioned by Mr. Mitereff. There are, however, instances as in the case of a sudden loss or of sudden surges in load when "higher-derivative"

[6] Hydraulic Engineer, C. J. Tagliabue Manufacturing Company, Brooklyn, N. Y. Mem. A.S.M.E.

[7] "Automatic Regulators, Their Theory and Applications," by Ed Smith, Jr., Trans. A.S.M.E., vol. 58, May, 1936, paper PRO-58-4, p. 291.

[8] "Controlling 1,500,000 Lb of Steam per Hour," by A. F. Schwendner, *Power Plant Engineering*, vol. 40, April, 1936, pp. 218–220.

control devices are highly desirable. We are not able to give the comparison between the calculated K_1 optimum and that obtained in practice at the present time. Mr. Mitereff assumes from our valve-characteristic curve, Fig. 2, that each following valve has to be opened prematurely, to be able to obtain such small deviations from the mean line. There is very little pressure drop allowed across any of the governing valves before the next valve is opened. The diffuser seat valves have the characteristics shown on the curve without resorting to a cam-type motion.

The authors welcome the suggestions and criticisms from the discussers and hope that the work done will encourage others to further contributions on the subject. Mr. Ed S. Smith, Jr., and his discussers in the February, 1937, issue of the A.S.M.E. Transactions, on "Automatic Regulators, Their Theory and Application," covered a related field quite thoroughly with a definite tendency to bring the regulating problem to a common ground by all concerned.

The Creep Curve and Stability of Steels at Constant Stress and Temperature[1]

C. Richard Soderberg.[2] The wealth of data upon which the author's conclusions have been based gives the above paper particular significance. Equation [1] of the paper appears well-established as a representative pattern for the variation in time of the plastic deformation in long-time creep tests at constant stress. Test series extending over 5 years is a new phenomenon in engineering science, and the author's firm is to be congratulated on its outstanding contribution in this particular direction.

The premise of a common time pattern for the creep curves of a given material forms the basic assumption of a companion paper[3] by the writer in which the subject is discussed from the point of view of design applications. In the latter paper[3] the assumption is also made that the creep curves at different stresses for the same material are geometrically similar. It is naturally of considerable interest to examine whether the data contained in the paper under discussion support this conclusion.

The data given in Tables 1 and 4 of the paper can be represented quite well by the equation $\epsilon = ST$, where S is a function of stress alone, and T is a function of time alone. Table 2 does not contain enough data for a satisfactory comparison. Table 3 gives rather large discrepancies for the highest stress where the calculated plastic strain is less than the actual. This is a common occurence which is undoubtedly caused, in part at least, by the reduction of area of the specimen. A plastic deformation of 18.4×10^{-3} (Table 3, $\sigma = 15{,}000$, $t = 5450$) represents a reduction of area (increase of stress) of 1.84 per cent. This will cause about 9.2 per cent increase in the creep rate for this material, which accounts at least for a part of the discrepancy.

On the whole, the agreement obtained by the author is closer than that obtained by the writer, except for the material in Table 4 of the paper, which from the point of view of the writer's method is not an "abnormal" material. It is possible to conclude, however, that the equation $\epsilon = ST$ is not an unreasonable simplification, particularly in view of the fact that it represents a necessary assumption in order to reduce the problems to manageable form. In the present state of the art, no solution of two- or three-dimensional problems can be expected, unless the phenomena at different points of the body can be assigned a common time variable.

When we come to the question of a physical interpretation of the expressions involved, the two papers[1,3] differ considerably. The author's arguments concerning metallurgical instability do not appear unreasonable, but the question is if all materials do not undergo a structural change, whether this change is clearly exhibited by microstructure or not. The answer to this question must depend on the quantitative definition of strain hardening. It is unfortunate that throughout our engineering literature this term has been left so vague that it can be made to mean any change from an ideal pattern of the flow.

With our present understanding of the phenomena of plastic flow in solids, it is natural to regard the viscous flow as a phenomenon in which there is no strain hardening and no structural change with time, these two influences being the only known causes of departure from the ideal viscous flow. The curvature of the strain-time function, defined by Andrade as plastic flow, is thus to be regarded as the combined influence of these two phenomena. If a quantitative evaluation is required, the second derivative ($\ddot{\epsilon}$) could be used; a negative sign denoting hardening and a positive sign softening. While not absolutely self-evident, it would be natural to suppose that this change of curvature is independent of the stress. The strain hardening must be some function of the plastic strain, however, and this fact should afford a means of at least a qualitative distinction between the two phenomena. On this basis the curvature due to strain hardening should increase with the strain, so that the curves for large strains and high stress ought to exhibit a more marked departure from the straight line than those for small strains and lower stress. It is a well-known observation from typical creep curves that this is definitely not the case. The writer has been led to believe from this reasoning that the phenomenon of strain hardening is insignificant in its effect as compared to the effects of structural changes. In this respect the fundamental outlook presented in the two papers[1,3] appears quite different.

If strain hardening is to be given a prominent place in the quantitative evaluation of the departure from the ideal form of viscous flow, it must be on the basis that a compensating structural change is assumed to depend on strain or stress, in addition to temperature and time, but this is merely logical subterfuge.

For the sake of logic it would seem desirable to use the terms, strain hardening and structural change, in the following sense:

Strain hardening: A change of the coefficient of viscosity demonstrated to be a function of strain only.

Structural change: A change of the coefficient of viscosity demonstrated to be a function of temperature and time only.

It is earnestly hoped that this important question be given further study in the future by metallurgists.

Author's Closure

The four sets of creep tests to represent four principal types of instability in steels are presented in the paper as Tables 1 to 4 inclusive. These tests ran for various periods from 5000 hr and longer the first set of four items has just completed its sixth year. Due to the size of the test curves they could not be presented in curve form for general distribution. Four 8 × 10-in. specially prepared photographic copies are filed in the archives of the A.S.M.E. and also in the Engineering Societies Library in unpublished papers for 1937 in order to facilitate photostatic reproduction. Complete results have also been filed with the Subcommittee on Data of the A.S.M.E.–A.S.T.M. Joint Research Committee on the Effect of Temperature on the Properties of Metals.

[1] Published as paper RP-58-16, by S. H. Weaver, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Manager, turbine engineering department, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa. Mem. A.S.M.E.

[3] "The Interpretation of Creep Tests for Machine Design," by C. Richard Soderberg, Trans. A.S.M.E., vol. 58, November, 1936, paper RP-58-15, pp. 733–743.

TABLE 1 CREEP RATES IN 10^{-7} IN. PER IN. OR PERCENTAGES PER 100,000 HR

Test item	V_{1000}	V_{Final}	Duration, hr
281	0.87	0.33	45,000
282	0.62	0.23	45,000
283	0.50	0.125	45,000
284	0.38	0.08	45,000
607	78.5	54.5	5,000
608	17.9	6.3	7,500
611	7.8	2.9	14,000
610	3.8	1.3	14,000
612	1.5	0.35	14,000
289	34.2	28.2	5,500
290	12.2	7.6	5,500
291	3.4	1.7	5,500
292	1.2	0.52	5,500
297	..	10.00	750
298	10.5	0.78	10,000
299	7.8	0.67	10,000
300	4.4	0.30	10,000

During the meeting, R. S. Brown requested the tested creep rate at the 1000-hr period for comparison with the rate at the end of each test. In Table 1 of this discussion are given these creep rates in 10^{-7} in. per in. per hr or percentages per 100,000 hr. An examination of this table is instructive as to the change of rate between the end of the 1000-hr period and the end of the test.

J. J. Kanter pointed out the author's inconsistency in the data for item 298 in Table 4 in the text. The calculated data given are for test points at 200, 400, and 700 hours with creeps of 2.26, 2.83, and 3.43 mils, respectively. This curve would pass through 1000 hr at 3.92 instead of 3.80 as given. The reason for using the 200, 400, and 700 hr in this case only, instead of the usual 200, 500, and 1000 hr, will be evident from the semilog plot of

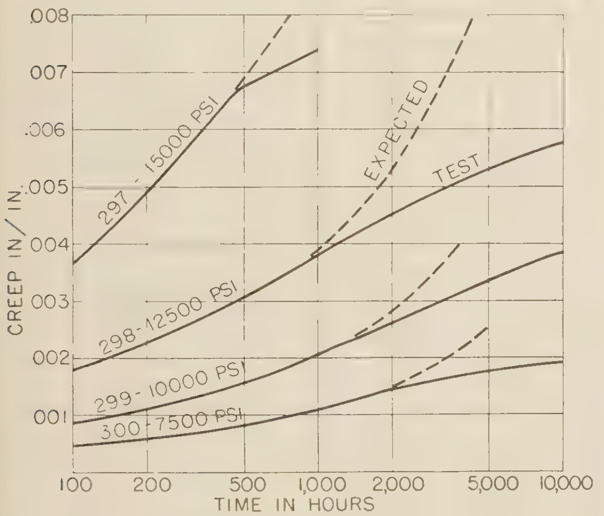

FIG. 1 CONSTANT-STRESS CREEP-TIME CURVES

the constant-stress curves given in Fig. 1 of this discussion. Item 297 is added for the fourth test item in the one furnace. On a semilog plot the constant-stress curves for a stable steel should have an increasing curvature and slope with the time as shown by the dashed lines marked "expected." Due to the great structural changes in this particular steel during the creep

TABLE 2 EXTENSION-TIME POINTS FROM FIG. 1 WHERE CURVATURE IN CREEP-TIME CURVES STOPS

Item	Stress	Creep	Hours
297	15,000	6.50	430
298	12,500	3.43	700
299	10,000	2.08	1000
300	7,500	1.21	1250

test, the curvatures cease to increase where the dashed curves leave the solid-line test curves. This occurs at the extension-time points listed in Table 2 of this discussion. Comparative creep constants should be derived for time shorter than that given in this table. For this reason the author used the special procedure for item 298 in Table 4 of the paper, then forgot to make a notation to that effect.

Fig. 9 of the paper also shows the advantage of the semilog plot for detecting unstable steels in creep. On a linear-scale plot, the curve variation from that for a stable steel is not easily observed.

The calculated creep curve for the additional item 297 is

$$\epsilon = 18.1 \times 10^{-7}t + 3.555 \times 10^{-3}\log t - 3.64 \times 10^{-3};\ t_0 = 10.6.$$

F. B. Foley stated that the ferritic banding in the rolled steel of Table 2 of the paper would be greatly strengthened and the banding "masked" by a quench-and-temper heat-treatment, then asked if the "masked" banding would be permanent or stable in long-time creep. By giving the annealed steel mentioned in Table 2 of the paper the proper heat-treatment for grain size and sorbitic structure, the banding is masked and the creep strength can be made to approach the high value of the steel in Table 1 of the paper. In this condition the steel is affected only by carbide spheroidization. The curves of Fig. 3 of the paper indicate that over 100 years at 842 F (450 C) is required to increase the constant creep rate to ten times the original value. This length of time is far longer than the useful life of any machine.

Mr. Soderberg's general equation for creep extension to equal a function of stress alone multiplied by a function of time alone is not invalidated when the equation did not apply to the very long-time creep tests given by the author. These tests were especially selected to illustrate four types of instability of steels during creep. They should not be used to check the accuracy of theoretical formulas as they are examples of the condition of metals which should be avoided in commercial use. This remark does not apply to the tests in Table 1 of the paper which agreed with Mr. Soderberg's equation. This test upon an unusually strong steel illustrated the effect of spheroidization of the carbides which could only be detected after two years of creep by measurements to the millionth of an inch per inch and in the sixth year of test the total creeps are 6 to 9 per cent greater than computed with the author's equation from the data obtained in the first 1000 hr.

There is no correction made in the tests of Table 3 of the paper for the stress increase due to the reduced sectional area resulting from creep extension. This test with Fig. 5 of the paper represents the effects of dendrites in steel castings when insufficiently heat-treated for good creep condition of cast steel. This steel is continuously strengthening over that indicated by the first 1000 hr of creep test and an area-stress correction would give a still greater change in the steel. The change within such a mixed structure cannot be easily reduced to a simple mathematical formula. The author is interested only in establishing the instability of dendritic cast steel when the dendrites are not sufficiently broken up by heat-treatment.

The definitions for strain hardening and for structural change in metals cannot, at the present time, be stated in simple mathematical terms, but must be descriptive of the multiple physical phenomenon involved in the plastic deformation of metals. The slow creep of steels over very long periods of time at elevated temperature has given additional weight to the time and temperature functions. It has been recognized for some time that the Ludwig formula for strain hardening does not extend over a broad enough field; therefore, we must be prepared to modify experimental formulas derived from yield-point plastic tests.

The metallurgist has advanced many theories for the hardening or strengthening of single crystals and polycrystalline metals. While the deformation characteristics of crystals or grains are

known, none of the hypotheses explain a quantitative theory for strengthening which fully accounts for both the plastic deformation and the strengthening. Excellent reviews of the present situation were recently given by Hoyt[4] for the single crystal and by Goss[5] for X-ray investigation of recrystallization.

The many able investigators mentioned in the foregoing references[4,5] have not included in their theories long-time creep with the important influence of time and temperature. Creep investigators have recently recorded the visible physical changes observed in metals during plastic flow. In lead at room temperature,[6] in iron and mild steel,[7] and in alloy steels[8] at elevated temperatures there has been found: (*a*) Grain distortion with elongation in direction of flow; (*b*) an occasional grain rotating as though relieving a localized strain; (*c*) an increase in grain-boundary material; and (*d*) slip planes through the body of the grains, the planes being inclined 45 deg to the tension stress. The slip planes are apparently part of the strain-hardening process, as they were observed in short-time tensile tests and in the first stage of a creep test, only to disappear when the creep became a minimum rate or constant rate and to reappear in the third-stage flow toward failure. Another characteristic (*e*) should be added where X-ray analysis gives the evidence (see paper by Goss,[5] pp. 981 and 1019). When plastically deformed steel is heated below the recrystallization temperature the X-ray diffraction lines become sharper due to a rearrangement of "fragmented" particles within the "blocks" between the parallel slip planes contained within the metallic grains. The result is a healing effect decreasing the hardness and strength.

Items (*a*), (*b*), and (*c*) can be the result of item (*d*), the slip planes being accumulated within the grains of metal and associated with the strain-hardening or strengthening gained by plastic distortion. Simultaneously, in the creep of steels at elevated temperature, the temperature is active in the healing or weakening of the fragmented parts of the grain. The creep of metals can be described as the result of two opposing effects. The creep extension is a strain-hardening or strengthening action while the elevated temperature produces an annealing or weakening phenomenon. When the rate of hardening from strain is equal to the rate of softening from anneal there results the constant-rate creep for a constant stress. This view of the effective strain-hardening would place the creep deformation as a function of stress, temperature, and time. For constant stress and temperature, the total creep due to strain hardening would be a time function only. The author assumed for this later condition that the instantaneous creep rate varied inversely with time and developed the creep formula in the Appendix of his paper.

Structural changes in metals vary in dimensions from the atomic-space lattice on up through the microscopic range to the large visible mixed grains of alloyed metals. Changes in the structure are usually accompanied by a permanent deformation which is a function of temperature, time, and even stress which can assist in the atomic movements. The author used the term "structural change" for any permanent change in the microstructure or structural formations of alloyed metals that is visible in the usual metallurgical microscope, and these structural changes, comparing the before-and-after creep test metal, furnished one of the means for classification of steels into stable or unstable condition for creep. The long-time tests with banding in rolled steel and with dendrites in cast steel were on materials with non-homogeneous structures where the large patterns formed by metals with different properties gave a mechanical effect which assisted in the erratic plastic distortions. Visible proof of structural change in these steels is difficult to obtain. The creep characteristics differ so widely from a steel of uniform structure that there was no hesitancy in classifying them as unstable.

The author would like to emphasize the utility of his creep formula in detecting steels which are unstable during creep. The check is really based on the fact that the tested creep curve on an unstable steel does not follow the formula. In routine calculations with the constant-stress tests the elongation-time curve is plotted on semilog paper and the formula applied to the points for 200, 500, and 1000 hr to determine the three constants in the equation. If $b > (a/2)$ the steel always has proved unstable. This means that the calculated curve is projected from the 200-hr point toward the origin and that zero time ($t_0 > 3$) is missed by more than 3 hr. This makes a very sensitive stability indicator for creep of metals below the equicohesive temperature.

[4] "Metallic Single Crystals and Plastic Deformation," by S. L. Hoyt, Trans. A.S.M., vol. 24, No. 4, December, 1936, pp. 789–830.

[5] "Hot Working, Cold Working and Recrystallization Structure," by N. P. Goss, Trans. A.S.M., vol. 24, No. 4, December, 1936, pp. 967–1020, particularly pp. 981 and 1012.

[6] "The Creep and Fracture of Lead and Lead Alloys," by H. F. Moore, B. B. Betty, and C. W. Dollins, Bulletin No. 272, February, 1935, Engineering Experiment Station, University of Illinois, Urbana, Ill.

[7] "Investigation of the Behavior of Metals Under Deformation at High Temperatures," by C. H. M. Jenkins and G. A. Mellor, *Journal of the Iron and Steel Institute*, vol. 35, No. 2, Sept. 18, 1935, pp. 179–236.

[8] "Influence of Time on Creep of Steels," by A. E. White, C. L. Clark, and R. L. Wilson, Proceedings of the American Society for Testing Materials, vol. 35, part 2, 1935, pp. 167–192.

Physical-Property Uniformity in Valve-Body Steel Castings[1]

V. T. Malcolm.[2] This paper is valuable because it reduces to understandable figures things which have existed in the minds of metallurgists and foundrymen in a very intangible and hazy form. However, the authors should not be too enthusiastic at this time in suggesting the possibility of substituting test bars taken from castings for separately cast test bars.

Steel foundry practice at its best is subject to variables which are, at the present time, somewhat beyond the foundryman's control. These variables are caused by the size and shape of the castings so that different parts and different areas must of necessity have different effects. Such being the case, some areas in castings are bound to be more difficult to form, and anticipated structures or strengths will fail to materialize in these unfavorable areas. In the present state of the art, it would seem much safer to proceed with caution, particularly in regard to the design factor in steel castings, because it must be remembered that, in testing steel, the steel bar itself gives an indication only of the quality of the metal.

Some years ago the writer, together with B. B. Wescott of the Gulf Refining Company, conducted an investigation over a period of several years on the subject under discussion, and later a summary of the investigation was published.[3] The complete report on this investigation is now being assembled and will be published in the near future.

The investigation as carried out consisted of the manufacture of steel in both the acid and basic electric furnaces, heat-treatment, heading and gating of steel castings, sand control, pouring practice. Approximately 30,000 lbs of steel were poured into

[1] Published as paper FSP-58-11, by A. E. White, C. L. Clark, and Sabin Crocker, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Director of Research, The Chapman Valve Manufacturing Company, Indian Orchard, Mass.

[3] "Specification for Chrome Tungsten Steel Castings," by B. B. Wescott, V. T. Malcolm, and H. Henderson, *Refiner and Natural Gasoline Manufacturer*, vol. 12, July, 1933, pp. 281–292.

special castings for this investigation. The castings were examined by X-rays and gamma rays, and sections from the castings were subjected to macroscopic and microscopic tests. The physical properties were determined from tension tests, Charpy impact, Eden-Foster repeated impact, as well as compression and torsional tests. A number of the castings, some of which were heat-treated, were subjected to destruction tests. In making a destruction test the casting was placed on a solid steel base and a 3000-lb ball was dropped on it from various heights until failure occurred. The investigations brought out the fact that there are several very important things necessary in the manufacture of steel valve castings. Among the most important the following may be listed:

Design of Valve-Body Castings, Headers, and Gates

It is necessary to design the casting in such a manner that thick and thin sections gradually merge without abruptness or sharp corners. It is also necessary to know the creep values of the metal entering into the valve body as well as the physical properties of the metal of like section. Therefore, test bars were designed that could be attached to the casting, and were of such dimensions that the sectional area of the attached bar corresponded with the greatest sectional area of the casting to which it was attached. However, it is of the greatest importance to know the values of the body itself, because slight flaws are sufficient to reduce the ductility and, therefore, its serviceability. In order to guard against this condition it is necessary to be thoroughly familiar with the laws of fluid pressure and to design headers and gates properly.

If castings are to be free from defects, and if it is to be determined whether or not the heading or gating has been correctly designed, it is necessary to conduct several tests, the most important of which are the X ray or gamma ray and the macroscopic etch.

In order to conduct these several tests, pilot castings must be made up before a new design is placed in production. These pilot castings must then be radiographed to determine whether or not the heads and gates will properly fulfill their function. Macroscopic tests are then made on pilot castings to determine segregates, if any, and crystal orientation. During the course of production, castings must be sent to the laboratory for radiographing and macroetching. Incidentally, the writer does not hesitate to accept specifications calling for these tests. Standard sets of radiographs and macrographs were developed in the investigation[3] previously referred to in order to determine the suitability of castings for the service intended, which includes service pressures and temperatures up to 1500 lb per sq in., and 1100 F, respectively.

Metallurgical control means control over materials entering into the composition of the steel, melting practice, teeming, sand conditioning, cores, heat-treating. In order to produce sound steel valve castings it is necessary that this control be exercised to the highest degree.

Heat-Treating—Metallographic Tests

Proper control with regard to heat treating is important, because otherwise satisfactory castings may be often rendered unfit for service by improper heat-treatment. It is therefore necessary to have properly designed furnaces, proper location of thermocouples and finally a heating and cooling cycle carefully determined for the type of steel and design of castings being treated.

The importance of effective heat-treating has been pointed out. The examination of the metallographic structure affords the most reliable means for evaluating results of the heat-treating operations, and may be so carried out as to place it in the class of nondestructive tests. The writer has found that the best way to make a metallographic examination is to remove a small section from a portion of the heaviest part of the casting in such a way as not to affect its usefulness. This test is believed by the writer to be a very reliable indication of the quality of the casting itself.

As a result of the investigation[3] previously referred to, the writer has concluded that the most satisfactory way of evaluating castings for high-pressure high-temperature service is by (1) properly designing test bars to determine the quality of the metal, (2) the use of radiographs and macrographs of pilot castings to determine soundness, (3) metallographic examination of sections taken from castings to determine efficiency of heat-treating, and (4) proper metallurgical control in the manufacture of the castings.

In closing this discussion, the writer wishes to state it is daily becoming plainer to those connected with the manufacture and application of castings for high-pressure high-temperature service that the present standard casting specifications are inadequate, and that if beneficial use is to be made of the data on the properties of metals at elevated temperatures, which are being assembled with such a high degree of precision, it will be necessary to use more rigid methods of inspection. These inspection methods can be made wholly practicable only by the closest cooperation between producers and users of castings. Thousands of castings are giving satisfactory service today, but failures from other than normal causes are of sufficient frequency as to point out the necessity for greater uniformity of quality.

Excellent alloys are available which are particularly suited for high-temperature high-pressure operation, either because of their unusual high creep strength or corrosion resistance, or both. However, if these alloys are to have any economic value they must assure a normal service life. Such assurance can only be had by the knowledge that the castings are integrally sound and properly heat-treated.

Therefore, no precaution is too great where it is possible that a single failure may cause loss of life and enormous property damage. Such jeopardy places a very high premium upon (1) actual knowledge of the property of metals that enter into the manufacture of castings and (2) the assurance of dependable quality of the castings themselves. Therefore, it is essential that the capabilities and the limitations of the castings be actually known so that safety of operation can be adequately assured.

R. A. Bull.[4] There are several phases of this paper which prompt a steel foundryman to form more or less definite impressions. The contribution is illuminating in respect to the physical properties found in the various portions of steel castings designed for use in any installation similar to that at the Conners Creek power plant.

The writer believes the paper supplies a very useful addition to the rather sparse existing data relative to the characteristics in all portions of valve castings. Occasionally this problem is of serious consequence to designers; particularly as they become more impressed by the fact that a conventional test coupon made in the regular way in the steel foundry cannot correctly be regarded as a criterion for the physical properties existing throughout all members of the commercial castings supposed to be represented. The coupon usually is and should be formed under conditions that closely approach the ideal from a practical metallurgical standpoint. This means that adequate provision should be made for amply feeding the coupon to compensate for the natural contraction of the steel as it cools in the sand mold. Furthermore, the coupon ordinarily is and should be formed under conditions that will offer safeguards against the entrapment of gases developed when the liquid steel comes in contact with the mold.

[4] Consultant on steel castings, Chicago, Ill.

Occasionally there are objections raised by some individuals who do not perhaps appreciate all the circumstances; leading to the argument that the specimens from all test coupons used as a basis for acceptance should be formed under conditions identical with those applicable to the commercial castings. Theoretically the argument may seem logical. But in so far as physical properties are concerned, the ordinary steel casting must be regarded as a composite of a number of members, each of which may have rather widely differing physical characteristics as the sole result of its design, and the opportunity thereby created for a good or bad degree of feeding to counteract the effect of shrinkage.

Since there are many members and portions of them which are necessarily affected by their opportunity to obtain metal from risers or sink-heads, it seems necessary to use a type of coupon for rejection purposes which is capable of standardization. That means a coupon made under all conditions which lend themselves best to uniformity. Unless we employ something that can be standardized, conclusions regarding the characteristics of the metal used for making the castings will be deceptive.

The coupon, made under conditions metallurgically ideal from a steelmaker's standpoint, has irrefutable arguments in its favor. But it is manifestly important that engineers, given certain responsibilities in connection with power and other plants, have supplementary means for ascertaining the probable behavior of the entire assemblage under consideration (such as a complete pipe line). Here the engineer may be fully justified in ascertaining the properties in all units employed. In making an effort to do this, he must appreciate the fact (obviously realized by the authors of the paper) that one cannot expect the characteristics in the conventionally made coupon to be similar to those obtained from specimens cut from any portion of every typical steel casting. The degree to which desirable properties may exist in the latter, as compared with the properties in the coupon formed under ideal conditions, gives the engineer a satisfactory indication of the probable serviceability of the installation.

The authors have thrown much needed light on what may be looked for in portions of steel castings of the particular types used in the Conners Creek power plant. Without discussing in detail the interesting comparisons drawn after comprehensive testing, it may be remarked that, on the whole, the steel-casting companies concerned did a very good job. And those who are associated with the steel-casting industry doubtless would agree with the writer that the authors have drawn very fair conclusions regarding the tests made on the products of the four foundries. The conclusions indicate an appreciation of the enormous influence of design on the properties in steel castings.

The writer is prompted to mention the fact that the design of the castings tested by the authors lent itself rather favorably to the development of satisfactory properties. This is because there is what might be termed almost a minimum of permissible inequalities in thickness, considering the intended application of the castings. Probably most mechanical engineers realize, as the result of information emphasized by steel foundrymen and confirmed by others, that radical inequalities of thickness represent an inherently defective condition in the steel casting. Furthermore, the efforts ingeniously made by experienced foundrymen to overcome the handicap of poor design are but partially successful in most cases. This is because of the high degree of contraction that is characteristic of steel, and because of the mass effect that develops conditions (as the result of slow cooling) which cause the thick section to have structural conditions frequently differing greatly from those found in a thin member of the same casting. It is a mistake to assume that all such differences in structure can be eliminated by heat-treatment. And of course the resistance to mechanical stress is greatly influenced by the granular structure.

In contrast with the design of the castings used for testing by the authors, the typical steel valve castings having a very heavy flange and an abruptly adjoining thin body wall is a very inferior composite of connected members. It is utterly unreasonable to expect that under the most expert conditions of foundry practice that can prevail, one may obtain identical physical properties in the thin body wall, in that connecting portion which joins the latter to the very heavy flange, and in the thick flange itself. The necessity for liberally feeding the heavy flange sometimes induces very great cooling stresses at the junction of the thin and thick portions; all to the ultimate disadvantage of the casting's resistance to mechanical stress, if great stress is to be exerted at the critical points.

It would appear that the growing practice of using welding fabrication for constructing pipe lines where components are made of cast steel has certain important advantages with respect to the better application of sound principles of design, in so far as the steel foundry is concerned. In the Conners Creek powei plant the typical thick flanges with which we are all familiar were not required. It would seem that the welding arc was here employed to the inherent advantage of the steel casting's resistance to stress. The writer has long believed, incidentally, that welding engineers and steel founders have a common ground for close cooperation, and that there is no logical reason for them to compete with each other in any spirit of unfriendliness.

Perhaps the only significant point in this discussion lies in an argument for better consideration by power-plant engineers and others to the vital influence of design on properties that can be developed in the steel casting. The nearer the approach to uniformity in cross-sectional dimensions which can be obtained while meeting assemblage requirements, the better will be the characteristics of the steel casting employed.

H. W. Maack.[5] The authors of this paper are to be complimented on their excellent presentation of the results of a thorough examination from the standpoint of the mechanical properties of castings of a steel now favored by engineers for pipe-line applications in steam power plants.

Study of the Charpy impact values of specimens from the four castings shows a quite wide range in the order of impact value for the different castings investigated, as well as in specimens from different parts of the same casting. The latter differences are probably due to differences in density or soundness in various parts of the valve-body castings. Specimens from casting *A* having the greatest density also had the highest impact value in the majority of cases.

The difference between the results on separately cast and integrally cast coupon tests are not accounted for so readily. A study of the microstructure of some of the test specimens with widely different results might account for the high and low impact values. The writer would like to ask the authors whether or not they made any such study. Besides density, the effectiveness of the heat-treatment in refining the grain is probably an influence.

With reference to the figures illustrating the macrostructure of the sections from different parts of the castings, inasmuch as methods for macroetching have not been standardized, it would add to the completeness of the paper if the details of the macroetching procedure were given. Concentration of the etching acid used, temperature of the acid and number of minutes immersion should be given. Most likely these detailed conditions were alike in all cases so that the illustrations are comparable in this respect.

Macrographs of etched sections from another valve-body casting of carbon-molybdenum cast steel are reproduced in Figs. 1 and 2 of this discussion representing, respectively, cope and drag sections from the bonnet flange and welding end of a 10-in. body.

[5] Chief Chemist and Metallurgist, Crane Company, Chicago, Ill.

These were etched for 45 min in a 50-50 muriatic-acid–water solution at 190 F. Except that the temperature is somewhat higher, these conditions are in accord with the recommendations in the 1936 handbook of the American Society for Metals.

Comparison of these macrographs with those of the paper show the former to be illustrative of a casting of soundness equal to the better sections shown by the authors. Such degree of freedom from imperfections is probably all that can reasonably be expected of castings of this size and design.

Fig. 1 Macrograph of Cope Section of Carbon-Molybdenum Cast-Steel Valve Body

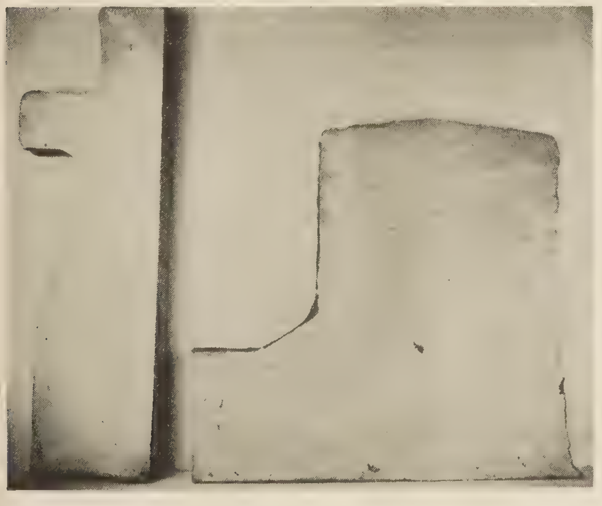

Fig. 2 Macrograph of Drag Section of Carbon-Molybdenum Cast-Steel Valve Body

The dendrites shown on the macrographs in Figs. 1 and 2 of this discussion indicate the manner of crystallization of the steel. This is influenced by pouring temperature, still or disturbed freezing, freedom from dissolved gases, and other conditions. Regardless of these influences and effects, the overall soundness of the casting and mechanical properties of the steel in various locations are the ultimate criterion of service usefulness. Suitable heat-treatment diffuses the carbides from the dendritic segregate, resulting in a uniform fine-grained microstructure which is largely responsible for the excellence of the mechanical properties.

In addition to the quite satisfactory degree of uniformity in physical properties from specimens in different parts of the valve-body castings examined, the data probably speak well for the general quality characteristics of carbon-molybdenum cast steel. Other low-alloyed steels might not give as favorable results, besides not being as satisfactory from the standpoint of welding qualities.

J. Roy Tanner.[6] The scope of the investigation made by the authors of this paper and the thoroughness and impartiality with which it has been conducted deserve great credit. The conclusions reached should be welcome information to those engineers who, through modern steam pressure-temperature trends, are being forced to accept responsibility for the successful performance of such equipment to be subjected to service more severe than any compassed by personal experience or observation.

It may be of interest to comment briefly on the fact that three out of the four castings coming from independent makers and of design differing in some details were quite uniform and practically acceptable under the specifications and to suggest an explanation of this close agreement in results.

Steel valves and fittings have been in the process of evolution since the turn of the century. They were at first made from existing patterns designed for iron castings and poured from the same ladle as the rest of the product of the jobbing steel foundry. Under such conditions the casualties on the test bench were appalling but in an industry whose product is repetitive such a condition can be, and, if the makers are to live, must be corrected. Any of the faults illustrated by the macrographs in the paper, if present to any degree, will inevitably produce leaks on test, thus adding greatly to production costs. Consequently self-interest, if nothing else, compels their reduction to a minimum.

Two features inherent in valve design have been of great help in locating the causes of unsound castings. Drilling the heavy flange sections at frequent intervals for bolt holes unfailingly uncovers piping due to shrinkage, and the machining operations necessary to seat the body castings are pretty sure to reveal defects in that unavoidably heavy locality. The detection of such faults has been a great help to the foundryman in improved feeding and gating which, together with the determination of proper pouring temperature, has greatly reduced failures on test; thus substantiating two of the authors' suggestions for improved product. The third, namely, proper heat-treatment, must be perfected along with sound metal if desired physical properties are to be attained.

Author's Closure

In commenting on Mr. Malcolm's discussion, the authors wish to disclaim the intention of advocating as routine test procedure the substitution of sections cut from actual castings in lieu of conventional test bars. The latter afford a reasonably satisfactory indication of what may be expected from proper casting conditions and represent an ideal which may be approached through proper design and good foundry procedure. The dissection of an occasional valve body for check tests or for verifying the merit or lack of merit of some new technique is a different matter, however, which the authors commend as well worth-while.

In view of the reasonable agreement between the properties of test bars and actual valve bodies as demonstrated in the paper, it is somewhat questionable if the use of the special test bar described by Mr. Malcolm possesses any special virtue.

[6] President, Pittsburgh Valve Foundry & Construction Company, Pittsburgh, Pa. Mem. A.S.M.E.

While the authors are heartily in accord with the practice of sectioning pilot castings in order to locate and correct regions of excessive porosity and to check the metal structure, they question the advisability of cutting out sections from regular production castings for macroetching. It is doubtful if this would be necessary as an inspection requirement where adequate control of heat-treatment and casting procedure has been developed.

Mr. Bull's comments on the desirability of using a standardized coupon, cast under ideal conditions, as a check on the characteristics of the metal proper, are quite apropos. As a matter of fact, the investigation was made to determine in what way coupons from different sections of the valve-body casting differ from these standard test coupons. The welding-end valve-body castings, while eliminating the heavy end flanges, still retain heavy bonnet flanges. Consequently, variations in thickness of adjoining members still exist in such castings. Future developments may permit the use of more uniform sections and bring about the condition implied by Mr. Bull as existing with present welding-end valves. But the heavy bonnet flanges now employed do not permit this ideal condition to be realized.

The authors have not attempted to correlate differences in impact values with the microstructure of the various impact specimens as suggested by Mr. Maack.

The details of the conditions of macroetching requested by Mr. Maack are as follows. All the specimens were etched at 160 F for approximately 1 hr in a solution consisting of one part of commercial hydrochloric acid to one part of water. These conditions compare with the procedure reported for the macrographs submitted by Mr. Maack of etching at 190 F for 45 min. in the same solution. The macrographs of sections from an additional valve-body casting submitted by Mr. Maack furnish confirmatory evidence of the general soundness of carbon-molybdenum valve-body castings.

The author's wish to thank Mr. Tanner for his considered comments on the evolution of steel castings and the importance of securing sound sections throughout, all of which tends to support their own views in the matter.

The Contact-Mixture Analogy Applied to Heat Transfer With Mixtures of Air and Water Vapor[1]

A. A. Markson.[2] While some of the physical ideas advanced in the author's paper would form the basis for several informal discussions, the body of the paper seems to form a highly useful statistical method of attack on the problems mentioned in the paper. Agreeing with the author's statement that he has formulated a useful analogy, the writer would like to see how the method is applied to the cooling-tower problem and how this compares with the cooling-potential method.

Author's Closure

Replying to Mr. Markson's discussion, the formula for the contact-mixture gives identical results with the cooling-potential method first formulated, it is believed, by Coffey and which the author judges is the method to which Mr. Markson refers. The cooling-potential method is, in itself, an analogy correlating the relationship of latent and sensible heat cooling. It is obviously not as generally applied to all problems as the contact-mixture method. It is also more remote from the true physical phenomena involved. The perfect agreement of the two methods, as far as results are concerned, however, may be checked either by psychrometric theory or by actual calculation of a specific problem.

The contact-mixture theory, however, takes into account an additional factor that the cooling-potential method does not, and that is the factor of air velocity. The cooling-potential method considers a comparison only of like velocities.

There may be some advantage in the contact-mixture method in the simplicity in calculation, especially where the psychrometric chart is employed. The contact factor, or the bypass factor, must be determined experimentally from a given form of cooling tower. It can, however, be approximated theoretically if the design of the tower is known. The observed performance will be found somewhat better than the theoretical, due to the fact that the coefficient of diffusion of vapor in the surface film is greater than the coefficient of the diffusion of air, as pointed out in the original paper, although for engineering calculations they can generally be assumed to be approximately the same.

[1] Published as paper PRO-50-2, by W. H. Carrier, in the January, 1937, issue of the A.S.M.E. Transactions.

[2] Assistant Research Engineer, New York Steam Corporation, New York, N. Y. Jun. A.S.M.E.

A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems[1]

David Dropkin.[2] This is a subject of great interest to engineers at the present time and Mr. Carrier is to be congratulated upon the research, both experimental and analytical, which he has conducted over a long period of time, relating to air and water vapor mixtures. Some of these relations are not too well understood at the present time.

This paper, however, raises some important points which are at least open to question. It must be remembered that in dealing with the difference between the wet-bulb temperature and the true temperature of adiabatic saturation we are often dealing with small fractions of a degree Fahrenheit and that therefore the utmost care must be exercised to get truly representative readings. Before undertaking the experiments reported in Bulletin No. 23 of the Cornell University Engineering Experiment Station,[3] referred to in the paper and from a modification of which one of the curves of the authors' Fig. 1 was drawn, a set of thermometers made to special specifications to fit this particular purpose were purchased and every care was exercised to obtain accuracy. An investigation was also made to determine the effects of different methods of applying the cloth covering to the wet-bulb thermometers. It was found that if the usual practice of covering only the bulb was followed, the resulting observed temperatures were too high due to conduction along the stem of the thermometer. A comparison of thermometers so covered with those having not only the bulb, but 8 in. of the stem covered and wet, showed that the readings of the thermometers with only the bulbs covered were 1.35 per cent of the wet-bulb depression higher than those having the more complete covering. This error is very large as compared with the 0.5 per cent correction, suggested by the authors, to apply to psychrometric observations. Using the first example given in the paper in which the actual wet-bulb temperature is 69.9 F, the velocity is 1200 fpm and the dry-bulb temperature is 90 F, the probable correction

[1] Published as paper PRO-59-1, by W. H. Carrier and C. O. Mackey, in the January, 1937, issue of the A.S.M.E. Transactions.

[2] Member of the research staff in mechanical engineering, Cornell University, Ithaca, N. Y.

[3] "The Deviation of the Actual Wet-Bulb Temperature From the Temperature of Adiabatic Saturation," by D. Dropkin, Engineering Experiment Station, Bulletin No. 23, July, 1936, Cornell University, Ithaca, N. Y.

when, adding the 0.5 per cent as the authors suggest, would be: 20(0.005) — 20(0.0135) = — 0.17 F. This gives 69.90 — 0.17 = 69.73 F as the adiabatic saturation temperature as compared with 70 F given in the example.

There also seems to be some confusion as to what air velocity the authors consider necessary in order that the wet-bulb observations will be identical with the temperature of adiabatic saturation. In the early part of the paper this velocity is located some place between 800 and 900 fpm, while in Appendix 2 it is given as between 500 and 600 fpm. There appears to be no evidence here that the value of slightly above 1000 fpm as stated in the previously mentioned Bulletin,[3] and substantiated by the modified Arnold theory, is not correct.

When a continuous duct having varying diameters is used to find the effect of velocity on the readings of the wet-bulb, and thermometers are placed along the duct in series, there is danger, unless the air is completely saturated, that the specific humidity will actually be slightly increased as it passes each thermometer. The upstream thermometer will give the true reading corresponding to the velocity; the downstream thermometer will give the reading corresponding to the initial air-vapor mixture plus the vapor absorbed from the preceding wet-bulb thermometer, and of course its corresponding velocity.

If the duct is well insulated, the vapor from the upstream wet-bulb thermometer causes a partial adiabatic saturation of the air flowing by it. This phenomenon is evidenced by the fact that the dry-bulb temperature at the downstream station is lower than that at the upstream station. Data from such an experimental setup yield a curve the slope of which is less than it should be. The following example makes the above statement clearer.

Suppose that the upstream velocities are lower than the downstream velocities, and suppose that the upstream wet-bulb thermometer should read 70 F and the downstream thermometer should read 69.95 F. It is known from the previously mentioned Bulletin,[3] that the actual wet-bulb temperature is lower than the temperature of adiabatic saturation at velocities above 1000 fpm. Therefore, were it not for the partial adiabatic saturation, the downstream wet-bulb thermometer should read lower than 69.95 F. Let us assume that this thermometer would read 69.90 F. Then, using these data for plotting a curve similar to the curve in Fig. 1 of the paper, we would have the increment as shown by the experimental data to be 70.00 — 69.95 = 0.05 F, while the correct increment should be 70.00 — 69.90 = 0.10 F. It is obvious that the first increment yields a curve with less slope than does the second.

In the Carrier-Lindsay experiments[4] this type of continuous duct with varying diameters was used and it is evident that they experienced this decrease in dry-bulb temperature after passing each station. The data given in Table 2 of this paper[4] show that in five out of the six sets of readings the observed dry-bulb temperatures at the downstream station were lower than those at the upstream station. Fig. 14 of this paper[4] shows the method which was used for finding the relation of wet-bulb temperature reading to air velocity. This method of arriving at results is subject to the error pointed out previously in this discussion, and it is reasonable to conclude that the curve in Fig. 1 of the paper[1] marked "Carrier-Lindsay Formula Test Results 1924" is lower than it should be at air velocities below 2000 fpm.

Authors' Closure

There are two points raised in Mr. Dropkin's discussion. First, there is undoubtedly an effect upon the reading of a wet-bulb thermometer due to the method of applying the wick to the bulb. Only the bulb of the wet-bulb thermometer of the usual sling psychrometer is covered, while the bulb and part of the stem were covered in the Dropkin experiments; for a given air velocity and state of the mixture, the wet-bulb thermometer of the sling psychrometer will read a slightly greater temperature than that obtained by Dropkin. This effect, alone, tends slightly to raise the authors' curves of Fig. 1 when they are applied to a sling psychrometer.

Mr. Dropkin's second point concerning the procedure followed in the Carrier-Lindsay experiments is not so well taken, however. Here he has assumed data which do not correspond to the tests. The point raised concerns the effect of the evaporation of water from a wet-bulb thermometer upon the reading of another wet-bulb thermometer placed downstream. In the tests to which Mr. Dropkin refers, wet- and dry-bulb thermometers were placed at two points of different cross-sectional area in the same air stream for the purpose of investigating the effect of air velocity upon the reading of the wet-bulb thermometer. In the run to which Dropkin specifically refers, the air velocity at the first section was 14.3 fpm, at the second section, 917 fpm, and the weight of dry air passing through the duct 210 lb per hr. The average wet-bulb temperature at the first station was 45.0 F and at the second station, 43.4 F; the decrease in wet-bulb temperature between the two stations was 1.6 F. The average dry-bulb temperatures at the two stations were 54.7 F and 54.5 F, respectively. From the heat balance of the wet-bulb thermometer, the weight of water evaporated, from the first wick, may be estimated, for, with symbols used in the paper

$$Mr' = \left(1 + \frac{h_r}{h_g}\right) h_g\,(t - t'),$$

or $M = \dfrac{1.406\ (1.95)\ (9.7)}{1068.4} = 0.025$ lb per hr per sq ft.

If the thermometer bulb and stem were covered for about 6 in. which was much more than that used in the test, the wet surface would be about 0.033 sq ft, so the weight of water vapor added to the air stream from this thermometer would be about 0.000825 lb per hr. This would cause an increase in specific humidity of the air of about 0.0275 gr per lb. This addition of water vapor occurs under substantially adiabatic conditions and represents a case of partial adiabatic saturation. The consequent decrease in dry-bulb temperature may be readily calculated as about 0.02 F. In other words, the calculated decrease in dry-bulb temperature at the second station, due to evaporation of water from the wet-bulb at the first, is 0.02 F. For the velocity that exists at the second station, the wet-bulb temperature is about 0.5 per cent of the wet-bulb depression lower than the temperature of adiabatic saturation. For a constant temperature of adiabatic saturation, then, the effect upon the reading of the wet-bulb thermometer, due to a change in dry-bulb temperature, may be expressed simply as

$$\Delta t' = -\frac{0.005}{0.995}\,\Delta t$$

$$= -0.00503\ \Delta t$$

A decrease in dry-bulb temperature at the second station of 0.02 F will cause an increase in the wet-bulb temperature of about 0.0001 F. The observed decrease in wet-bulb temperature between the two stations is 1.6 F. Failure to consider the effect of evaporation from the first bulb upon the reading of the second wet-bulb thermometer introduces an error of about 0.006 per cent. Most engineers would consider this an insignificant error. The authors, therefore, see no reason for changing the curve of

[4] "The Temperature of Evaporation of Water into Air," by W. H. Carrier and D. C. Lindsay, Trans. A.S.M.E., vol. 46, 1924, p. 739.

Fig. 1 which was based, in part, upon the Carrier-Lindsay experiments.

Regarding the air velocity at which the wet-bulb temperature is the same as the temperature of adiabatic saturation for mixtures of water vapor and air, the authors believe the experimental data to be insufficient at the present time to fix this, precisely. Rather than fixing this velocity at 1000 fpm, exactly, they would prefer to suggest that it is somewhere between 300 and 1000 fpm. At an air velocity of 750 fpm, Dropkin's tests give the difference between the wet-bulb temperature and the temperature of adiabatic saturation, expressed as a per cent of the wet-bulb depression, as -0.32, -0.14, $+0.96$, $+1.12$, and $+1.59$. If more weight be assigned to the lower readings, for the reason that there may well be incomplete wetting at the highest depressions, Dropkin's data will prove this point.

Tests of a 50,000-Sq Ft Surface Condenser at Widely Varying Temperatures, Velocities of Inlet Water, and Loads[1]

George C. Eaton.[2] The author has chosen to neglect throughout his tests and calculations the effect of dirtiness of the condenser tubes on the resistance of the tube wall to heat flow between the condensing steam and the circulating water because of the fact that during all the test runs the circulating water was treated with chlorine. At the Edgar Station of the Edison Electric Illuminating Company of Boston, it was found during experimental work with chlorination that condenser performance was adversely affected by mud or silt deposits on the bottom inside surface of the tubes, even though all traces of slime were absent. The author was indeed fortunate if, due to chlorination, it was not necessary to clean the tubes of his condenser at intervals throughout his test period, even though the time spacing of such cleanings was several times greater with chlorination than without it.

Three years ago the writer prepared a set of curves similar to those shown in Fig. 16 of the paper from over-all heat-transfer coefficients expected by the condenser manufacturer during commercial operation of the apparatus. These curves are used to compare actual condenser performance and serve to determine the proper cleaning intervals. The writer's curves are for a constant inlet-water temperature and show the relation between the condensing steam temperature (plotted as ordinates) and condenser steam flow (plotted as abscissas). The curves were of course based on the previously mentioned over-all heat-transfer coefficients and a constant flow of circulating water, to which flow the circulating-water pumps are adjusted prior to a routine test. Such curves as shown in Fig. 16 of the paper, or those described by the writer, appeals to the writer far more than ones involving the relation between the Reynolds number and the ratio of initial-temperature difference to internal heat loading suggested by the author in his conclusions to "establish the all-year-around performance of a condenser."

The author limits his paper to "the parts of the steam-air circuit outside the tube banks" and specifically states that "the hydraulic circuit is not treated." The writer, together with Robert E. Dillon of the Edison Electric Illuminating Company of Boston, and H. Peters of the Massachusetts Institute of Technology, has been studying the hydraulic circuit very intensively during the past year particularly as it affects the life of condenser tubes. The results of this study have recently been compiled in a paper, "The Prevention of Surface-Condenser-Tube Failures," for presentation at the Semi-Annual Meeting of the Society held in May, 1937, in Detroit.

R. A. Bowman.[3] The data and arguments presented by the author in support of the theory that the superheating of the condensate is caused by the inlet velocity of the steam going to the bottom of the condenser are quite conclusive. This theory seems to be the only reasonable one that has ever been advanced to explain this phenomenon, and it is gratifying to see that it has been corroborated quantitatively. It is interesting to note that, while the superheat amounts to 6 deg maximum at a cooling-water temperature of 33 F, for a water temperature of 62 F, which is probably close to the design point of the system, the superheat is a maximum of 1 deg and for water at 75 F it is not more than 0.4 deg.

Most design and operating engineers are accustomed to evaluating the performance of a condenser in terms of a heat-transfer rate. Consequently, would not the use of heat-transfer rates by the author, in place of resistivity, have made the author's ideas a little easier to grasp by the reader? The use of resistivity has a very decided advantage in that the individual resistances can be added directly, but for the sake of a common language, the use of heat-transfer rates outweighs the advantage. By the same reasoning, it would seem preferable to base the transfer figures on the external tube surface rather than on the internal surface, since condensers are ordinarily rated in terms of external surface and, again, that is the unit in which we are accustomed to think. There is no fundamental difference between rates based on the inside surface and those based on the outside surface, since they are different only by a constant multiplier, the magnitude of which depends on the tube thickness and diameter.

The use of dimensionless quantities, such as Reynolds' number, in tabulating heat-transfer data is highly desirable and is, of course, quite sound theoretically. In those cases where results given by several different fluids are to be correlated, the use of dimensionless factors is imperative. In the case of the surface condenser, when there is only one type of fluid to be considered inside the tubes, its use seems to complicate unduly the design and performance calculations. However, in the case at hand, there is some justification for its use since it permits correlating all the data in a single curve, so that general deductions can be made. As pointed out by the author, this is only permissible because of the fact that changing temperatures affect the Prandtl number only slightly.

The use of the condensate temperature, in place of the saturation temperature corresponding to the pressure at the condenser neck, for calculating the mean temperature difference of the exchanger is rather difficult to justify. It seems to the writer that there can be only two valid reasons for changing from the present practice of using the saturation temperature at the condenser neck. These are as follows:

1 That the new figure would give a more nearly correct average temperature difference, so that the heat-transfer rates as figured would be more nearly the correct values.

2 That the new figure would be more truly indicative of the back pressure against which the turbine is called upon to exhaust.

The temperature of the condensate, as pointed out, will ordinarily be somewhat higher than the temperature at the condenser neck, while the true pressure or temperature average

[1] Published as paper FSP-58-10, by G. H. Van Hengel, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Head of the Mechanical Technical Engineering Division, Generating Department, The Edison Electric Illuminating Company of Boston, Boston, Mass. Jun. A.S.M.E.

[3] Design Engineer, Westinghouse Electric & Manufacturing Company, South Philadelphia, Pa. Jun. A.S.M.E.

hroughout the tube bundle will very likely be lower than the emperature at the condenser neck, due to pressure drop through he tubes. Thus, it would seem that while the temperature dif- erence as at present calculated is beyond question in error, he use of the condensate temperature would also be in error and, f anything, the results would be even farther from the true state f affairs. Under this system, the apparent mean temperature lifference would be even higher than at present, and the apparent eat-transfer rates would be lower, making the difference between pparent and actual heat-transfer rates on the individual tubes ven larger. A comparison of Figs. 11 and 12 of the paper shows hat the results obtained by using a conventional logarithmic nean temperature difference are more consistent than if con- ensate temperature were substituted, and approach more losely the curves of Fig. 10, both in magnitude and direction.

The purpose of a condenser is, naturally, to produce a partial acuum into which a turbine exhausts. An accurate indicator f a condenser's performance is, then, one which is affected only by he vacuum in question. The author points out that the con- ensate temperature is affected, independently of the vacuum y the condenser design. Consequently its accuracy as an indi- ator of vacuum is open to question. After all, the primary urpose of a condenser is to produce high vacua at the turbine xhaust, not to deliver hot water from its condensate well. Consequently, based on the foregoing, the use of condensate tem- erature as an indicator of the condenser performance offers no dvantages either as a basis for calculations or guarantees.

While the use of a so-called inlet resistivity, as outlined in the econd part of the author's conclusion, gives rather good agree- nent between the various test points attained on this condenser, s indicated in Fig. 17 of the paper, there is some probability hat its use would be a cause of error in those cases where the emperature rise of the water is larger, as compared to the inlet- emperature difference. This condition would, of course, be btained in condensers having two or more passes or having maller and longer tubes. That this is true is indicated in Fig. 17 f the paper by the fact that the short individual lines do not arallel the average curve. This system might prove satisfactory n any given condenser, but the writer would hesitate to use it as general design curve.

Kenneth F. Wicks.[4] This report covers more detailed data nd theory concerning what actually happens inside of a surface ondenser than any paper the writer has ever read. Many apers have presented excellent theoretical information or ery good data from actual tests, but the close combination and terweaving of them have been lacking to a large extent.

Since the Hudson Avenue Station has all of the condensers ounted in line with the turbogenerator axes rather than trans- erse to them, the information contained in this report will have be modified before it can be compared directly. However, the onclusions as made in the report are quite general in character nd applicable to any installation. Naturally the effect of the elocity and direction of the steam at the entrance to the con- enser would be quite different.

The problem of obtaining suitable vacua readings for turbine rformance, and also the total pressure head, in order to elimi- te artificial readings of hotwell superheat has been quite oroughly covered. It is quite apparent that there can be no tual superheating. In a well-designed, properly laned conden- r, the hot-well-condensate temperature may be close enough to e true temperature under which heat transfer takes place, but similar figure for an unlaned condenser would be far from the uth. It is apparent, therefore, that for condenser performance the only true basis for computation is the actual total head of the steam entering the condenser. The problem then is one of obtaining a vacuum reading which will give the total pressure head. With this as a basis the computation should be nearer the truth.

The writer has suggested the use of resistivity rather than heat transfer as a method of measuring condenser efficiency. While this method is not entirely new, the author has presented his data in a rather different form than has been used in the past, and has simplified the computation as far as possible without losing the fundamental basis upon which it is built.

In short, while this report is extremely interesting in itself, it can only be of use as adapted to conditions for the particular installations to be studied. The true value, therefore, rests with the number of those in the industry who are sufficiently interested to apply it to their own peculiar problems.

The possibility of maintaining copper-constantan thermo- couples within an accuracy of 0.1 F is worthy of comment. While calibrations may possibly be repeated within these limits, it would seem quite unlikely that under test conditions such a calibration could be relied upon. The usual limits of accuracy applied even to a noble thermocouple are broader than these.

The possibility of exposure to a vacuum changing the calibra- tion of a mercury-in-glass thermometer seems quite remote also. Some further discussion of the reasons for this and proof that the thermometer and not the calibrating thermocouple was af- fected seems quite necessary. It seems highly improbable that either method of temperature measurement should be affected by such exposure.

D. W. R. Morgan.[5] The design of the condenser referred to by the author incorporates a well-known principle, that is, pro- viding a steam-filled zone through which the condensate must fall, thereby preventing undercooling of condensate without re- sorting to any complicated or restricted devices to produce the result.

Referring to the superheating of condensate, the writer recalls that similar results were obtained on a 32,000-sq ft condenser, the extent of superheat amounting to 2.5 deg; in another instance of a 16,000-sq ft condenser, the superheated condensate amounted to 4.5 deg. However, condensers are used primarily for reducing the back pressure to a minimum at the turbine exhaust, and the efficiency of dissipating this heat from the turbine exhaust to the circulating water should not be depreciated for the sole pur- pose of increasing the condensate temperature. On the other hand, it is desirable to withdraw the condensate at the highest temperature practicable. The method proposed by the author introduces the possibility of overloading the cooling surface within close proximity of the air-cooler section. This is un- desirable, since it decreases the efficiency of the condenser.

The value of increased condensate temperature should not be overestimated. For example, in an operating condition based on a pressure of 300 lb per sq in., a vacuum of 29 in. and a Rankine cycle efficiency of 80 per cent, 1 deg is worth 0.08 per cent in a straight-condensing turbine without bleeding of steam; 0.043 per cent when one stage of heating is employed, and 0.025 per cent if three-stage heating is used.

The author suggests that the temperature of condensate be used, instead of the temperature corresponding to the pressure at the steam inlet of the condenser, for evaluating performance. It would be unfortunate if such a method were adopted because the condensate temperature is variable depending upon the load and pressure within the condenser as well as the temperature of the inlet water. For example, the author shows a 6-deg differ-

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[5] Condenser Division Engineer, Westinghouse Electric & Manu- facturing Company, South Philadelphia, Pa.

ential with an inlet-water temperature of 33 F, a 1-deg difference with inlet-water temperature of 62 F, and a 4-deg difference with an inlet-water temperature of 75 F.

John F. Grace.[6] The condenser which is the subject of the paper under discussion is the No. 13 unit at the Delray Station of the Detroit Edison Company. Referring to the values for test No. 65 given in Table 1 of the paper, it is noted that the static pressure at the neck of the condenser inlet corresponding to the temperature (items 27 and 28), averages 0.396 in. of mercury. The hotwell pressure as given by item 13 is 0.548 in. of mercury, which is 0.152 in. higher than the static pressure at the inlet. Item 66 shows a velocity head of 0.0453 in. at the steam inlet, thus indicating that the velocity head is 0.1067 in. higher than the total of the static and velocity pressure at the inlet neck, but not as high as the total at the twenty-first stage.

In the next to the last paragraph on page 633, the author states: ". . . . . the total pressure of a flowing fluid never rises along the path of flow." This is correct when it refers to the entire mass, and it is possible that the apparent discrepancies shown in Fig. 5 of the paper may be due to local exhaust-steam inlet-velocity heads which are higher than those indicated by the average velocity, as surmised by the author. It is also probable that the hot-well pressure recoveries of three times the inlet-neck velocity head, as reported by the author, is the attainment of an objective sought in this Detroit condenser. This objective was the accumulation of pressure upon the smaller mass in the hotwell by surrender of kinetic energy in the form of a multitude of rapid impacts from the somewhat turbulent flow of the larger mass which turns from the central steam inlet lane into the tube banks.

From the Worthington (J. F. Grace[6]) specification filed November 19, 1936, U. S. Patent 1,855,231, the writer quotes: ". . . . . the present invention comprises a steam condenser of the surface type embodying a steam lane leading to the hotwell which lane is so shaped and proportioned that during the passage of steam therefrom, the velocity and kinetic energy of the steam will be converted into pressure and temperature or heat for raising the temperature of the condensate entering the hotwell by the transfer of said converted energy, as heat, to the condensate."

Author's Closure

Referring to Mr. Eaton's remarks about neglecting the dirtiness, the author intentionally made no corrections for dirtiness. This in itself is a wide field for investigation which is beyond the scope of this paper. The future will show that this correction for dirtiness includes also the phenomena of tube effectiveness. The author, however, claimed that the changes of dirtiness throughout the period of testing were negligible (see p. 634, second column, first paragraph). Without these corrections for changes in dirtiness, it was far simpler to attain the objective of this paper, i.e., a single-line representation of an all-year-around performance.

The assumption of negligible changes of dirtiness can be proved by Fig. 15 of the paper and also by some additional information from station operation records. In Fig. 15, substantially the same performance is shown for runs ten months apart which lie nearly in the same range of circulating-water inlet temperatures and Reynolds' numbers. Any appreciable change of dirtiness over the ten-month period of testing should have caused the performance of the 33-F runs, which were made at the end of the test period in January, 1936, to be noticeably poorer than the performance of the 42-F runs, which were made at the beginning of the test period in March, 1935. Fig. 15 shows that the 33-F runs and the 42-F runs are practically overlapping.

The most recent cleaning of the condenser before the start of the tests on March 26, 1935, had been done 10 months before, on June 2, 1934. The next cleaning took place on May 18, 1936, four months after the finish of the test; and this was done not because of increase in dirtiness, but as a routine operation during the overhauling of the main turbine. Between these cleanings, the condenser was in service 7038 hours, and on the basis of operating log data the improvement in back pressure due to the tube cleaning of May, 1936, was less than 0.01 in. Hg.

Comparatively favorable conditions of circulating-water supply as well as chlorination contributed to the author's good fortune in this respect; the usual unfavorable conditions of alluvial rivers, or of brackish or salt water with tidal flow, are not encountered because the circulating water of the Delray Station is taken from the Detroit River.

Fig. 17 of the paper gives in a single curve the all-year-around performance from test results. From this curve any line condition as in Fig. 16 can be calculated, assuming a certain inlet-water temperature and any circulating-water quantity. Mr. Eaton, however, replaces Fig. 17 by the manufacturer's data. As far as the author knows, the method of taking Fig. 16 for a rapid check of the performance of condensers has been commonly used for many years.

The author's recently published paper on the analysis and tests on hydraulic circuits of surface condensers[7] is a supplement to this paper which was too lengthy to include the hydraulic circuit.

In his second paragraph, Mr. Bowman says that the resistivity is not the logical thing to use. This may be true for those who in regular routine work use the over-all heat-transfer coefficient to calculate condenser performances, but for those who want to penetrate more deeply into the mysteries of a condenser, it is difficult to justify. For instance, plot on the curves of resistivity for the entire condenser versus Reynolds' number, such as Figs. 11, 13, and 15 of the paper, the resistivity of the single tube as found in Fig. 10, and we can visualize directly the effect of the so-called "dirtiness or cleanliness," because the difference between those resistivities shows the increase. This vision would be far more obscured if we should use the heat-transfer coefficient.

As the author pointed out at the bottom of page 635 of the published paper, 70 per cent of the total resistance to heat transfer in condensers is at the inside of the tube, and the inside surface is more important than the outside surface. This holds true especially when comparing condensers with different tube sizes, since the ratio of external to internal surface changes with different tube sizes (outside diameters) and different thicknesses of tube wall.

Because of the lack of data, the author's paper did not depart from the old practice of choosing a measurement at a single location as a criterion on which to base the performance of the condenser, in spite of the inherent fallacies of the conventional procedure. Scientifically, this is of course not justifiable, and for a comparison of the resistivity of a condenser with the over-all thermal resistivity of a single tube, as shown in Fig. 10 of the paper, the author would suggest basing the resistivity not on a single criterion, but on a weighted average of total pressure and heat loading. A properly weighted average of temperature and heat loading would take into account the temperature and available surfaces at all main points in the condenser, and it is a function of temperature and velocities available at the condenser neck, hotwell, air-cooler entrance, and air-cooler outlet.

Before going into the details of the author's choice of the condensate temperature, instead of the saturation temperature cor-

[6] Condenser Engineer, Worthington Pump & Machinery Corp., Harrison, N. J. Mem. A.S.M.E.

[7] "Analysis and Tests on Hydraulic Circuits of Surface Condensers," by G. H. Van Hengel, Trans. A.S.M.E., vol. 59, April, 1937, paper FSP-59-3, pp. 151–160.

responding to the static pressure, for evaluation of the all-year-around condenser performance, the author wishes to make the following statements concerning the turbine and temperature readings inside the condenser. These statements will clarify the misconceptions of this point in both Mr. Bowman's and Mr. Morgan's discussion.

The efficiency of a turbine depends on the actual work removed from the steam by the blades of that turbine. Since the exhaust losses (that is, the steam velocity energy from the last wheel of a turbine) are of no value for delivery of work to the turbine shaft, the total pressure at exhaust is the criterion by which to judge a turbine. A certain drop of total pressure takes place in the exhaust hood owing to the 90-deg change in direction of flow. This total-pressure drop subtracted from the total pressure at the turbine wheel gives the total pressure at the condenser neck. Just as it is the total pressure and not the static pressure which is of value to the turbine, the total pressure at the condenser neck is the criterion by which to judge the condenser performance since the temperature equivalent of the steam velocity energy at the condenser neck is available to the condenser. At present, the standard practice for calculating the heat-transfer coefficient is to use the saturation temperature corresponding to the static pressure at the wall of the condenser neck. This choice is erroneous because the impact on the tubes due to the steam velocity will raise the temperature of the steam around the tube.[8,9,10]

In a modern condenser, the static pressure will gradually increase from the condenser neck to the hotwell, due to diminishing velocity pressure. During a winter test on No. 3 main condenser at the Trenton Channel powerhouse[11] the author observed that the saturation temperature, corresponding to the static pressure at the top of the entrance to the air cooler, lay between the temperature of the steam, 10 in. in from the wall at the condenser neck, and the condensate temperature. This result was all the more unexpected because the pressure measurement was taken under the roof of the air cooler, which apparently would be more or less in a low-pressure region. The phenomenon of having a higher pressure in the hotwell *only* is therefore not true.

On the basis of the foregoing information, it is logical to consider the saturation temperature corresponding to the *total* pressure at the condenser neck as the temperature from which the heat transfer should start.

A representative value of the total pressure over the large area of the condenser neck was difficult to measure, and was not measured in the test discussed in the paper.

If a measurement at only a single point were to be selected as a criterion on which to base the performance of a condenser, as was done to avoid the difficulties of the measurement just mentioned, the author preferred to take the condensate temperature rather than the condenser-neck wall pressure measurement because: (1) An accurate measurement of it is more easily obtained; and (2) it is more closely related to the total pressure to which the turbine efficiency should be referred.

The last two paragraphs of Mr. Bowman's discussion and the last paragraph of Mr. Morgan's should therefore be conceived not in the light of the static pressure as the fundamental criterion, but the total pressure as the fundamental criterion.

[8] "Bestimmung von Wärmeübergangszahlen durch diffusionsversuche," by W. Lohrisch, *Forschungsarbeiten*, vol. 322, 1929, pp. 61–64.

[9] "The Mechanism of Heat Transmission: Distribution of Heat Flow About the Circumference of a Pipe in a Stream of Fluid," I, by T. B. Drew and W. P. Ryan, Transactions, American Institute of Chemical Engineers, vol. 26, 1931, p. 141, fig. 19.

[10] "The Temperature Field in an Air Current Flowing Across a Hot Cylinder," by J. Small, *Philosophical Magazine*, Series 7, vol. 19, 1935, p. 32, fig. 8.

[11] "The Trenton Channel Station of the Detroit Edison Company," *Engineering*, vol. 125, June 1, 1928, p. 670, fig. 113–116, inclusive.

In his last paragraph, Mr. Bowman is in agreement with the author's assumption stated in the paper that this application does not hold for every condenser or for a high circulating-water temperature rise. In the second paragraph on page 638 of the published paper, the author mentions this; the sentence, however, could have been made clearer by adding a restriction which was erroneously omitted, viz., "Since the temperature rise of the cooling water is low in a single-pass condenser *in comparison to the initial temperature difference*, the mean heat loading of the whole condenser may be substituted for the heat loading at the cooling-water entrance in calculating the resistivity."

Answering Mr. Wicks as to the accuracy of the thermocouples, it may be mentioned that in the field the same type-K potentiometer and all details of the setup were the same as used in the laboratory calibration test. Normal and reversed readings were taken.

The checks in the field were made in two ways: (1) By comparing the circulating inlet-water west thermocouple with a mercury thermometer; and (2) by comparing the hotwell-steam thermocouple with the saturation temperature corresponding to the pressure measurement of this steam made with the absolute-pressure oil gage. For a few of the most constant runs the hotwell

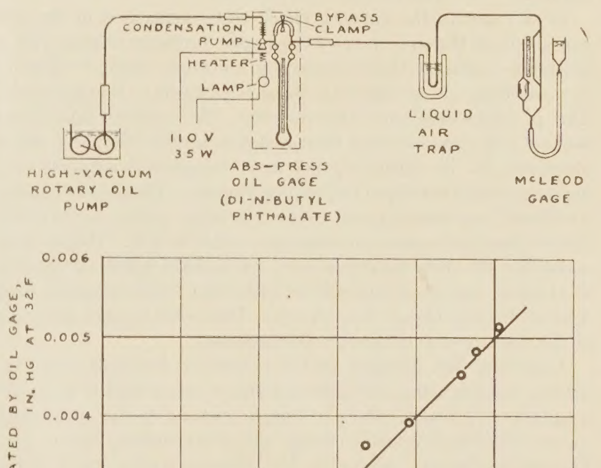

Fig. 1—Check of a Di-N-Butyl Phthalate Absolute-Pressure Manometer by a Comparison With a McLeod Gage

[For this comparison, the McLeod gage with its liquid air trap was connected directly to the high-pressure leg of the absolute-pressure oil gage by a short piece of rubber tubing. The entire system had been evacuated of air for 15 hr by leaving open the oil-gage by-pass clamp with the rotary oil pump (which was connected to the gage reference leg) operating, and also with the oil-gage condensation pump operating. The readings were made after closing the by-pass clamp of the oil gage, the pressure in the closed-off part of the system rising because of its disconnection from the evacuating system. The pressure rise from 0.000 in. Hg to 0.005 in. Hg covered a period of $2^1/_2$ hr. Footnote (13) of the paper gives more detailed information on the absolute-pressure oil gage, and contains a bibliography with references to the McLeod gage and high-vacuum technique involved in the setup of the high-vacuum reference system of the absolute-pressure oil gage]

comparison was shown in Table 6 of the paper; an example of a few of the circulating inlet-water temperature comparisons was given for the same runs in Table 3 of the paper.

The sensitivity and accuracy of the absolute-pressure oil gage filled with di-n-butyl phthalate is far superior to any type of mercury gage and is demonstrated by Fig. 1 of this discussion, which shows graphically the comparison of readings of one of the oil gages with a McLeod gage over the range of the latter, which was from 0 to 130 microns Hg (0 to 0.00512 in. Hg). At the highest pressure reading of the McLeod gage, 128 microns Hg, the oil-gage reading was a differential of 0.17 cm di-n-butyl phthalate at 79 F; so, at a pressure shown by the McLeod gage of 0.0050 in. Hg at 32 F, the oil-gage indicated a pressure of 0.0051 in. Hg at 32 F.

As to Mr. Wick's doubts of the mercury-thermometer vacuum-exposure correction, such correction is to be expected since the mercury thermometer is based on the difference in expansion of the glass bulb and the mercury contained in it. The thermometer reading will be affected by any changes in externally-imposed pressures which affect the expansion of the bulb (see the A.S.M.E. Code on Instruments and Apparatus, Part 3—Temperature Measurement; Chapter 6—Liquid-in-Glass Thermometers, paragraph 21—External Pressure Correction).

In addition to the vacuum check tests mentioned in the last paragraph of the appendix of the paper, previous checks with a platinum-resistance thermometer had shown no vacuum effect on the readings of an exposed copper-constantan thermocouple. The platinum-resistance thermometer, the thermocouple, and some of the same mercury thermometers used in the later check mentioned in the author's paper were compared when immersed in an evacuated insulated tank containing air. The thermocouple, a different one from the one used in the later checks, agreed with the platinum-resistance thermometer within 0.03 F. The vacuum correction for these thermometers, which have bulbs of Corning G-81 glass, agreed within 0.05 F with the correction established later on for the Delray No. 13 test. This is within the accuracy of the reading of the mercury thermometer.

Answering Mr. Morgan on the matter of designing for condensate superheating, the author wishes to state that it is simply a question of heating the condensate instead of the circulating water with steam velocity energy which the turbine cannot use. The thermodynamic saving, as Mr. Morgan points out, is not a large one, but is still worth-while, particularly as no extra investment is required to obtain it.

The answer to Mr. Morgan's last paragraph has been given already in the reply to Mr. Bowman's discussion, viz., that in case one adheres to a single criterion from which to judge a condenser, the total pressure at the condenser neck should be taken as the initial temperature for any condenser. In a condenser which shows reheating of the condensate, the condensate temperature comes the closest to the total pressure, and is therefore suggested for use.

In answer to Mr. Grace, the author wishes to state that the principle of using the exhaust-steam velocity to raise the pressure in the hotwell by providing a steam lane from the condenser neck to the hotwell is a very old one. However, the higher pressure in the lower part of the condenser is useful only if the condensate is afforded the opportunity to reheat.

The distribution of the steam velocity at the condenser neck will depend on the passage between the turbine-blade exhaust and the condenser, but the steam lane should probably be located where the highest velocities occur at the condenser neck.

All condensers such as those with the tube bundle placed eccentric toward the head end of the turbine, radical-flow condensers and two-lobe condensers will give the steam an opportunity to enter the hotwell.

The author applied this principle of using the exhaust-steam velocity when in 1930 he changed some condensers for his company, among which was No. 4 main condenser in Conners Creek powerhouse.[12] The great undercooling of the condensate prior to the 1930 change was practically done away with. The test results, though not of the precision of the author's tests since 1933, when the oil manometers were first used, showed a pressure rise from condenser neck to hot well of seven times the condenser-neck velocity head calculated on a basis of uniform flow over that area. The author made radical changes of a similar nature on three condensers in Trenton Channel in the summer of 1931. Tests made in 1934, of precision comparable with the Delray No. 13 tests discussed in the paper, showed that the pressure rise was three times the mean velocity head at the condenser neck. This result is practically the same as that for the Delray No. 13 Condenser shown in Fig. 5 of the paper.

Whether the steam lane should be placed in the center or on the side will in such a case depend on the steam flow in the passage between turbine-blade exhaust and condenser. This flow will therefore depend on such things as (1) the area of the last wheel; (2) how far the turbine-barrel casing extends into the turbine-exhaust hood; (3) the shape and length of the connection between turbine and condenser; and (4) the layout of the tube bundle in the condenser.

[12] Edison Electric Institute Publication B3, June, 1934, p. 3, Figs. 1 and 2.